United States Patent [19]

Leet

[11] 4,309,917

[45] Jan. 12, 1982

[54] **TRANSMISSION AND CONTROL SYSTEM**

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[73] Assignee: **General Motors Corporation**, Detroit, Mich.

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[51] **Int. Cl.**[3] **B60K 41/06; F16H 47/04**

[52] **U.S. Cl.** ..... **74/861**; 74/856; 74/687; 74/752 C; 74/869

[58] **Field of Search** ..... 74/856, 857, 859, 860, 74/861, 862, 869, 752 C, 687

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3,640,157	2/1972	Schaeffer	74/869
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3,752,010	8/1973	Tipping	74/687
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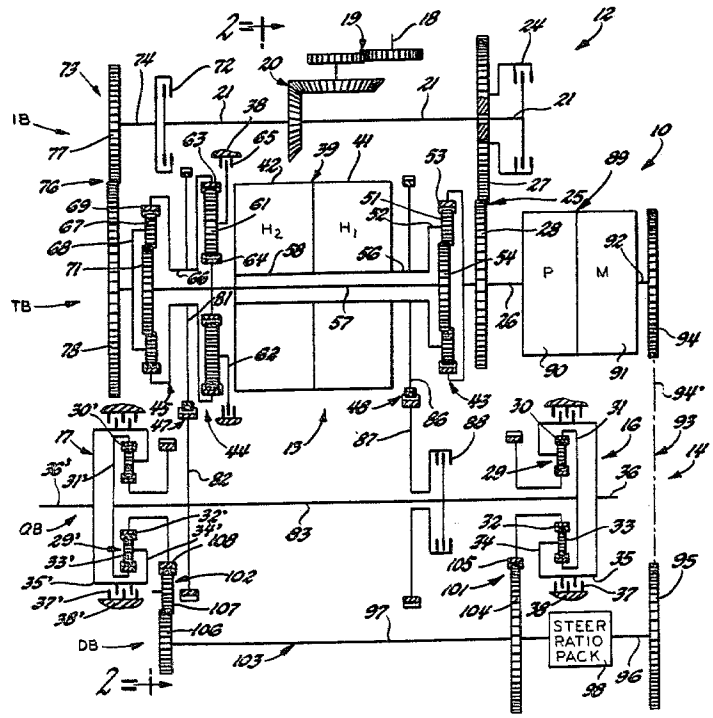
Primary Examiner—Leslie Braun

Attorney, Agent, or Firm—Saul Schwartz

[57] **ABSTRACT**

A cross drive transmission having a hydromechanical propulsion drive and hydrostatic steer drive. In the controls, a ratio governor has a set speed control wherein set speed is established in accordance with throttle position, steer request, engine power and brake torque. Advance and return signals regulate the speed ratio of the hydromechanical drive so input speed equals the set speed. A rate of change control provides fast increase of the return signal, slow increase of the advance signal in higher ratios and very slow increase of the advance signal in low ratio. Manual controls, return and advance signal control valves, shift signal valves and shift valves, in neutral, provide substantially zero displacement in the hydrostatic unit and neutral in the gear unit, and, in drive positions, coordinate hydrostatic ratio control and gear ratio control for continuous variable ratio change in the hydromechanical propulsion drive. Overpressure in the propulsion hydrostatic unit controls the ratio governor to provide only a return signal and disengages all gear ratio drives until output speed is reduced to a low value and hydrostatic unit displacement is reduced substantially to zero for smooth reengagement of drive. Underpressure in the hydrostatic steer drive disables the steer controls. The steer controls are also disabled at low engine speeds and low engine power so steer load does not stall the engine and in neutral manual range to prevent inadvertent steering.

6 Claims, 10 Drawing Figures



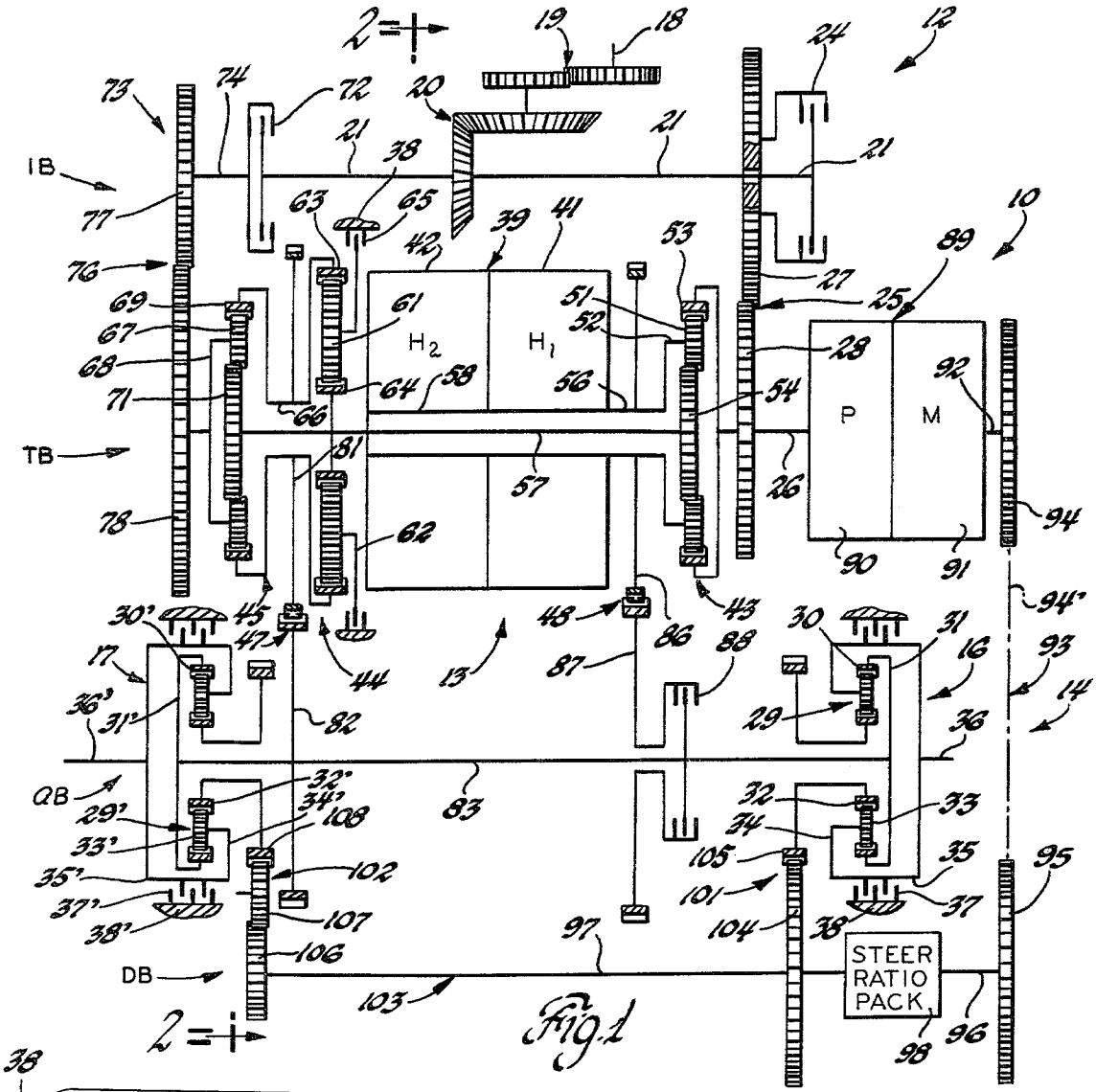


Fig. 1

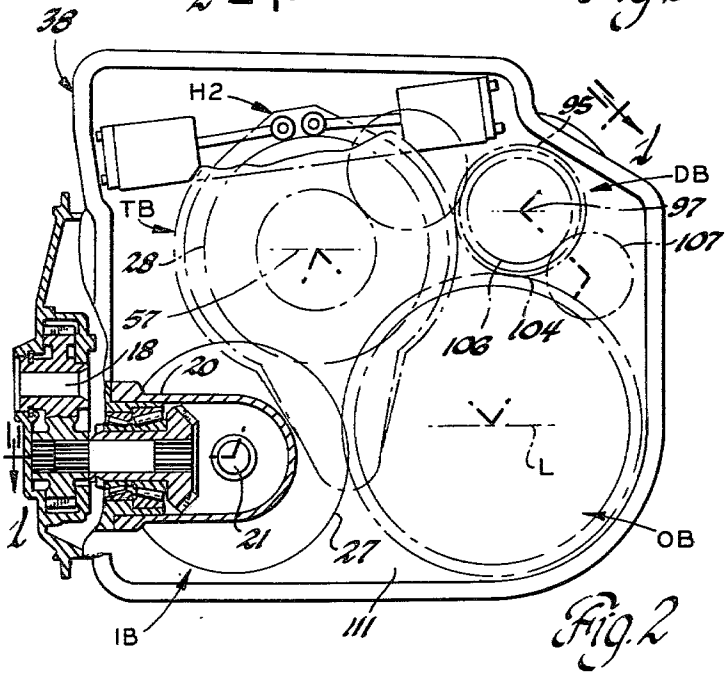


Fig. 2

Fig 3A	Fig 3B	Fig 3C
Fig 3D	Fig 3E	Fig 3F

Fig. 3

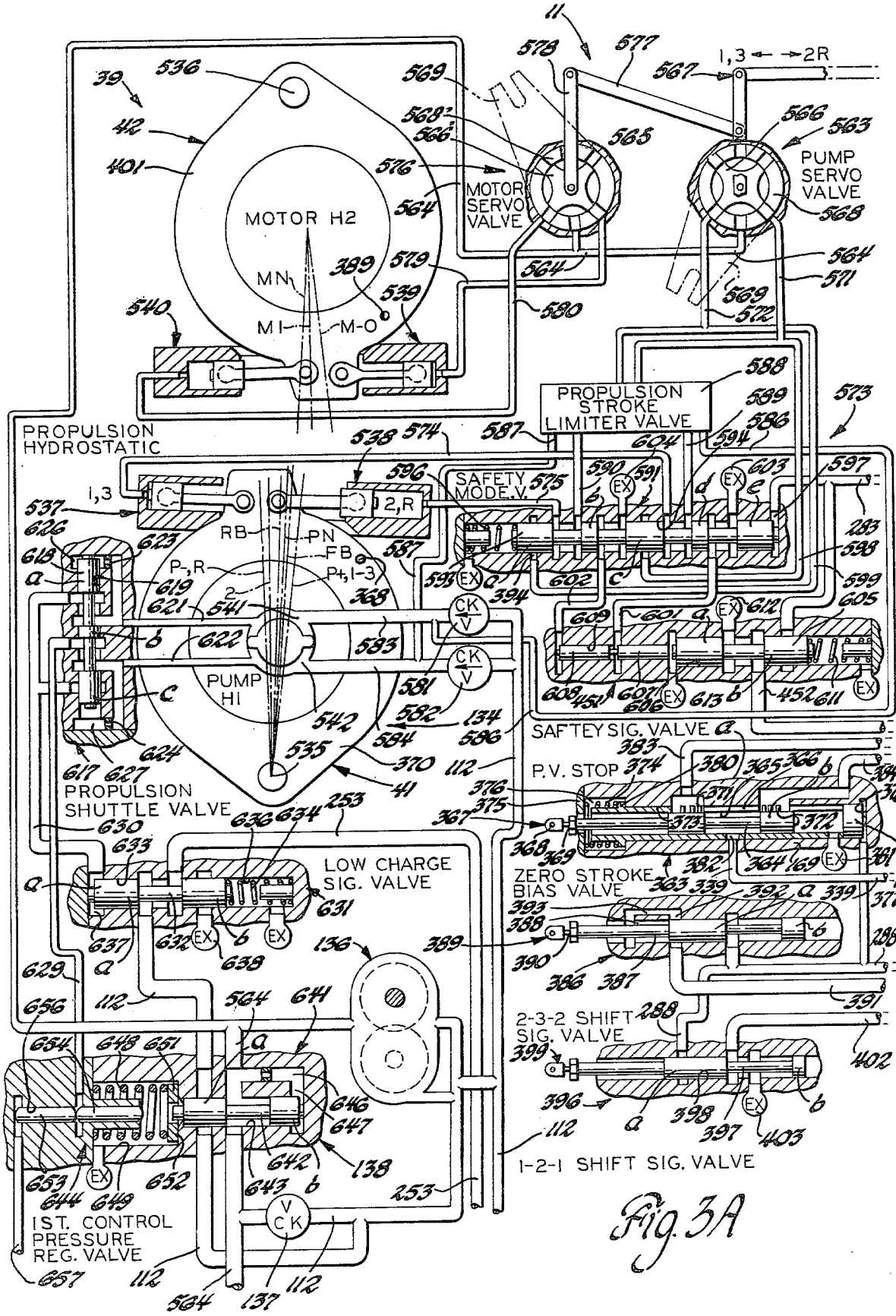

PROPULSION HYDROSTATIC
MOTOR H2
MN
MI
M-O
MOTOR SERVO VALVE
PUMP SERVO VALVE
1,3 ← → 2R
PROPULSION STROKE LIMITER VALVE
SAFETY MODE V.
PUMP HI
RB
PN
FB
P-R
P-1-3
PROPULSION SHUTTLE VALVE
LOW CHARGE SIG. VALVE
SAFTEY SIG. VALVE
P.V. STOP
ZERO STROKE BIAS VALVE
2-3-2 SHIFT SIG. VALVE
1-2-1 SHIFT SIG. VALVE
IST. CONTROL PRESSURE REG. VALVE
CK V
EX

Fig. 3A

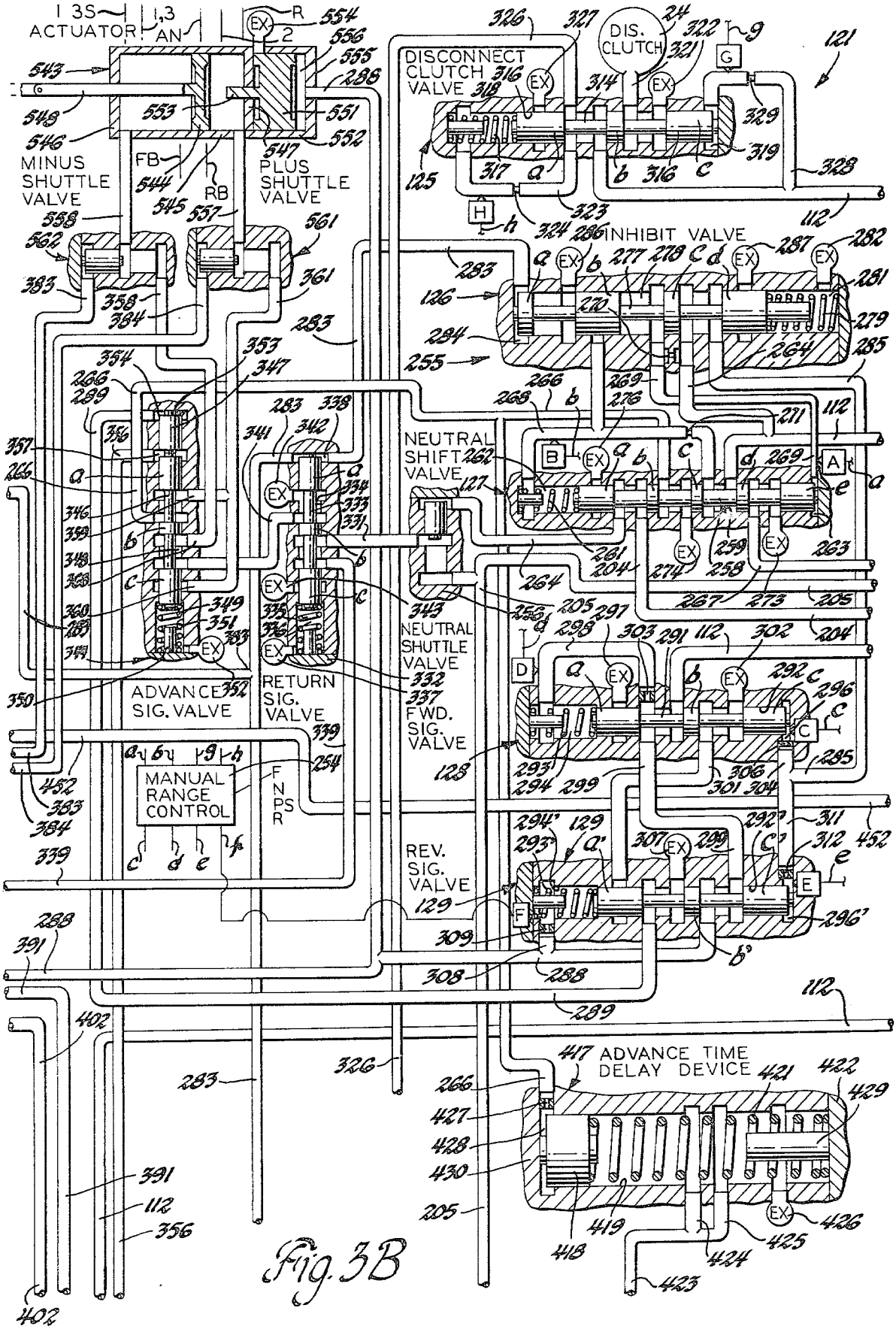

1 3S ACTUATOR
1,3 AN
DISCONNECT CLUTCH VALVE
DIS. CLUTCH
INHIBIT VALVE
MINUS SHUTTLE VALVE
PLUS SHUTTLE VALVE
NEUTRAL SHIFT VALVE
NEUTRAL SHUTTLE VALVE
FWD. SIG. VALVE
ADVANCE SIG. VALVE
RETURN SIG. VALVE
MANUAL RANGE CONTROL
REV. SIG. VALVE
ADVANCE TIME DELAY DEVICE
EX
FB
RB

Fig. 3B

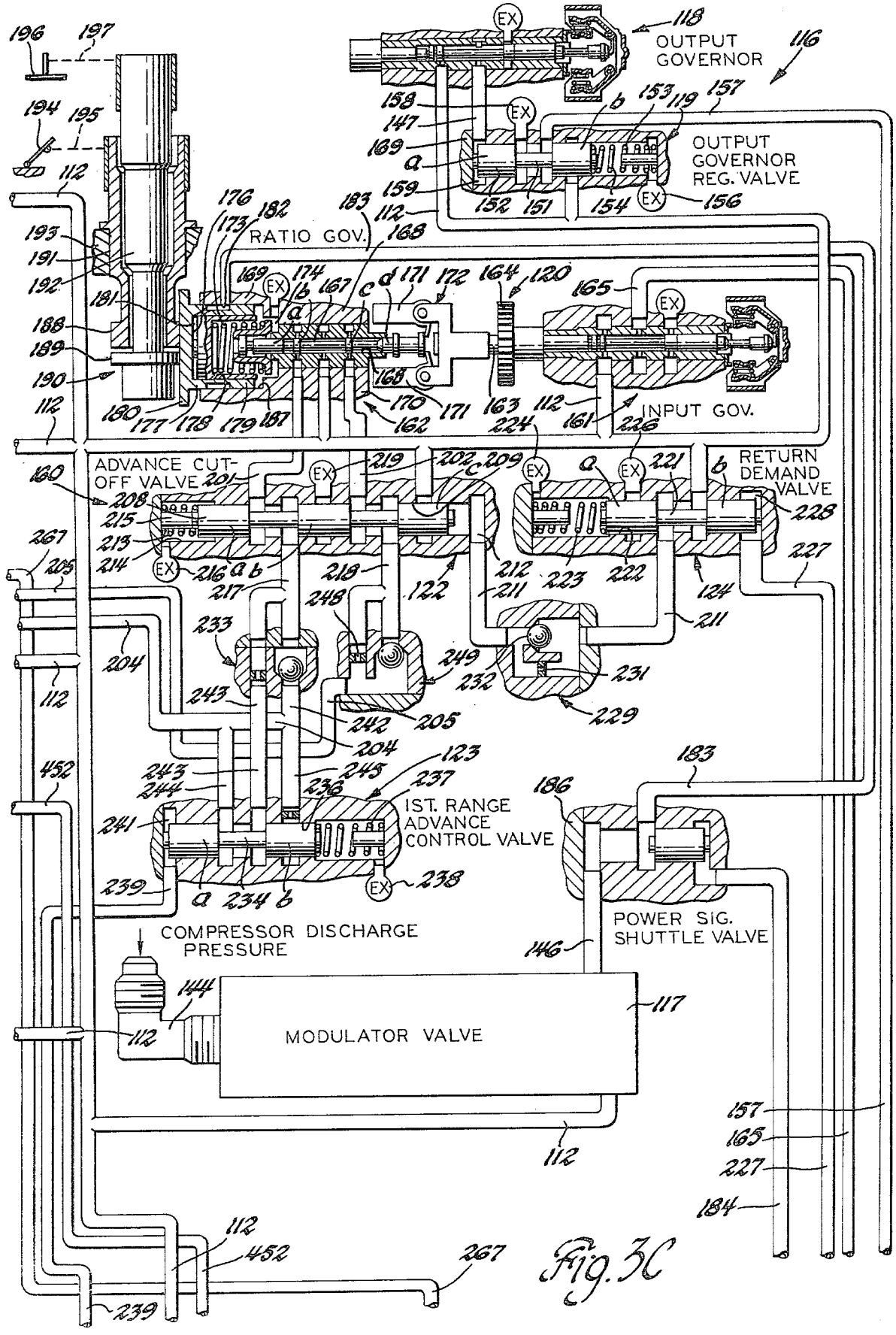

OUTPUT GOVERNOR
OUTPUT GOVERNOR REG. VALVE
RATIO GOV.
INPUT GOV.
ADVANCE CUT-OFF VALVE
RETURN DEMAND VALVE
1ST. RANGE ADVANCE CONTROL VALVE
POWER SIG. SHUTTLE VALVE
COMPRESSOR DISCHARGE PRESSURE
MODULATOR VALVE
EX

Fig. 3C

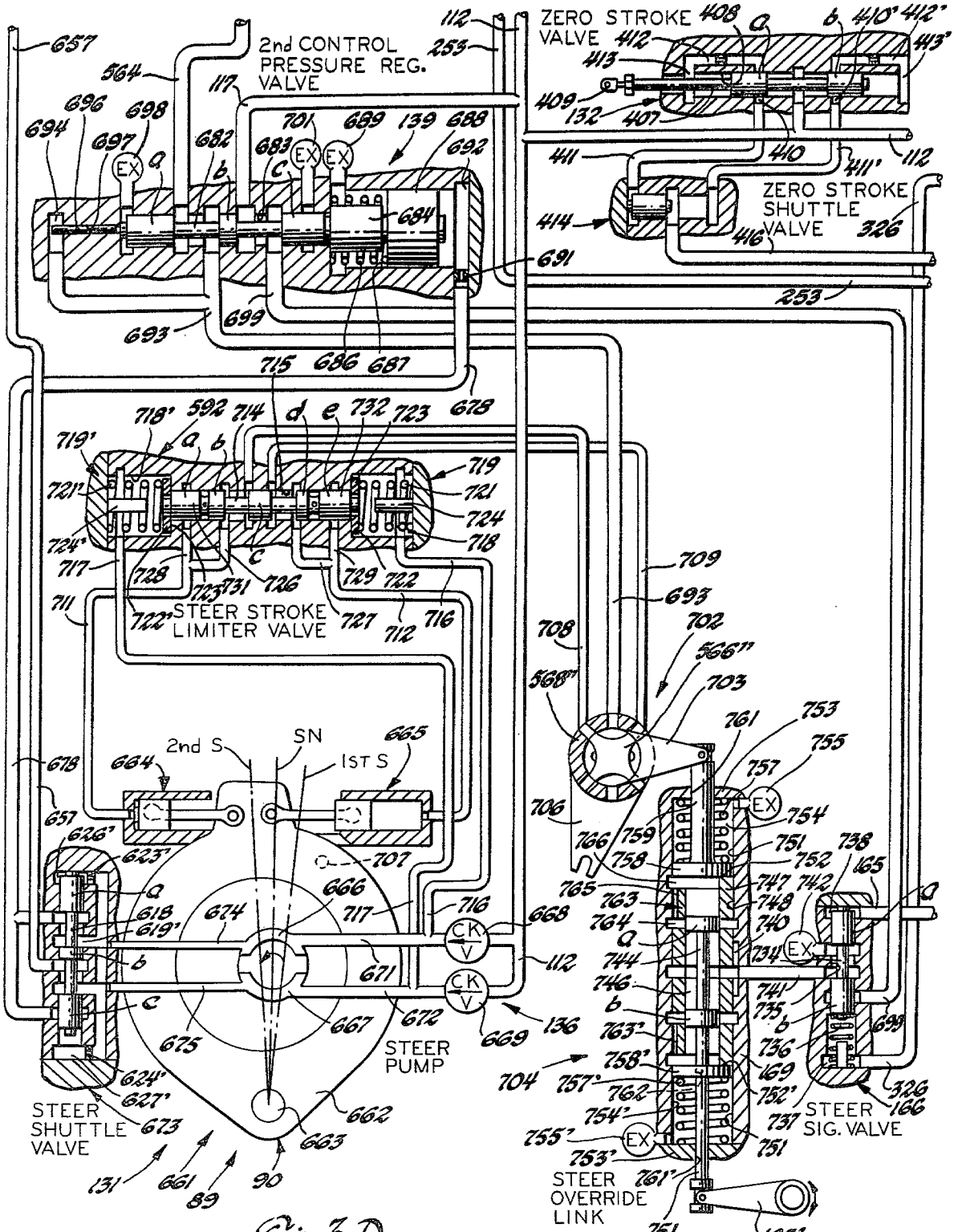

2nd CONTROL PRESSURE REG. VALVE
ZERO STROKE VALVE
ZERO STROKE SHUTTLE VALVE
STEER STROKE LIMITER VALVE
STEER PUMP
STEER SHUTTLE VALVE
STEER SIG. VALVE
STEER OVERRIDE LINK
2nd S
SN
1ST S
CK V
EX

Fig. 3D

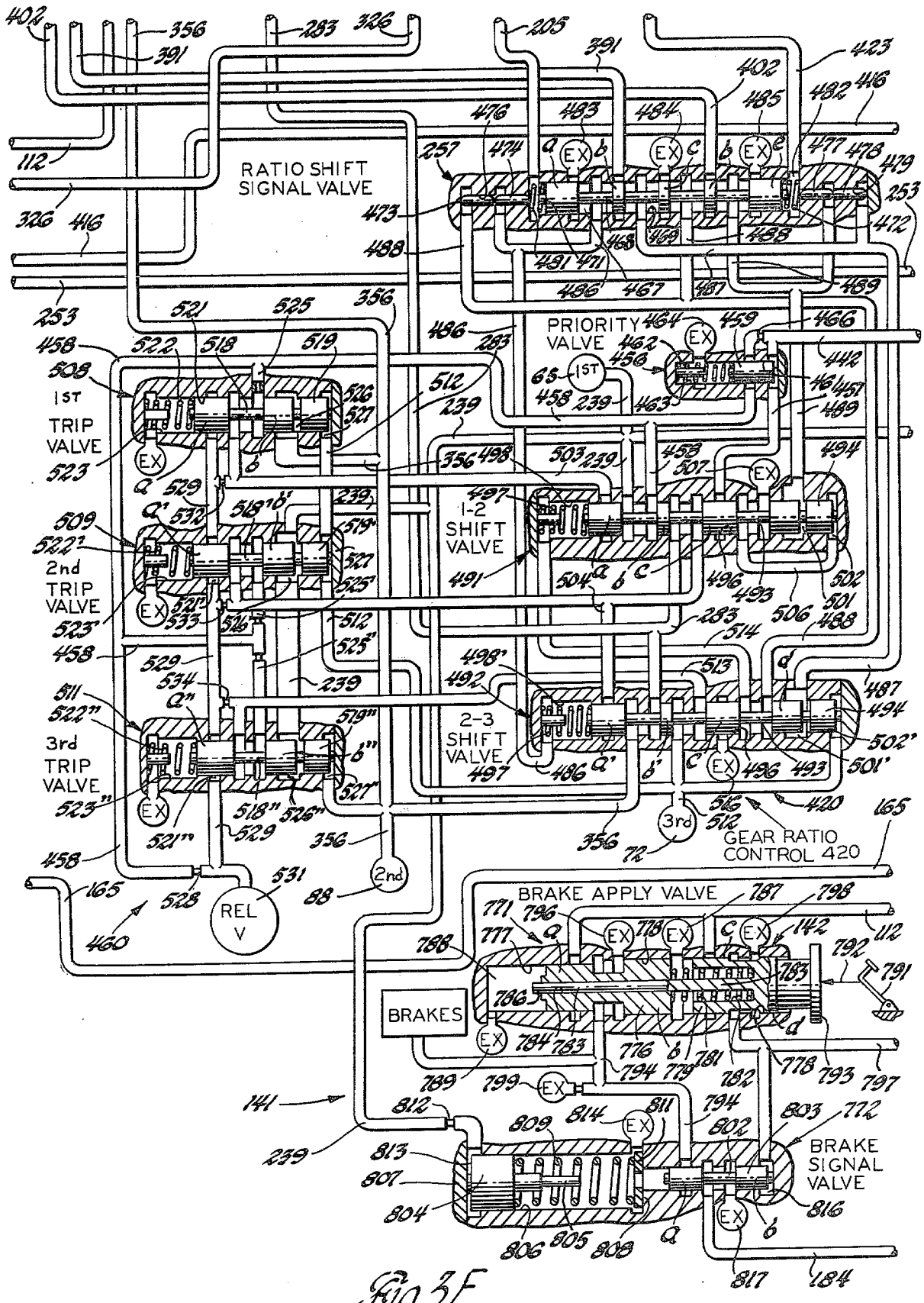

RATIO SHIFT SIGNAL VALVE
PRIORITY VALVE
1ST TRIP VALVE
2nd TRIP VALVE
3rd TRIP VALVE
1-2 SHIFT VALVE
2-3 SHIFT VALVE
GEAR RATIO CONTROL 420
BRAKE APPLY VALVE
BRAKE SIGNAL VALVE
BRAKES
REL V
1ST
2nd
3rd
EX

Fig 3E

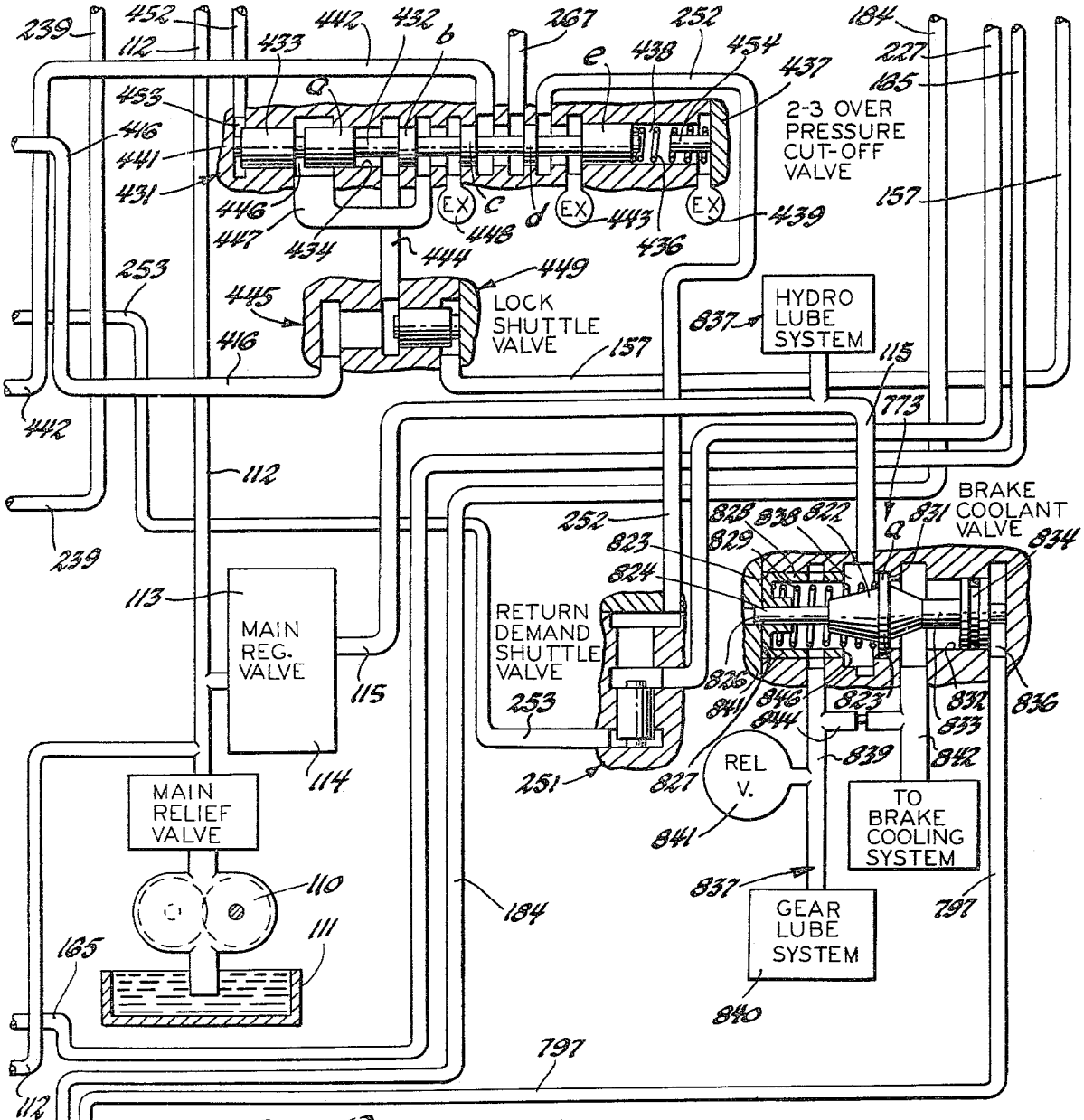

2-3 OVER PRESSURE CUT-OFF VALVE
LOCK SHUTTLE VALVE
HYDRO LUBE SYSTEM
BRAKE COOLANT VALVE
RETURN DEMAND SHUTTLE VALVE
MAIN REG. VALVE
MAIN RELIEF VALVE
REL V.
TO BRAKE COOLING SYSTEM
GEAR LUBE SYSTEM
EX
a
b
c
d
e

Fig. 3F

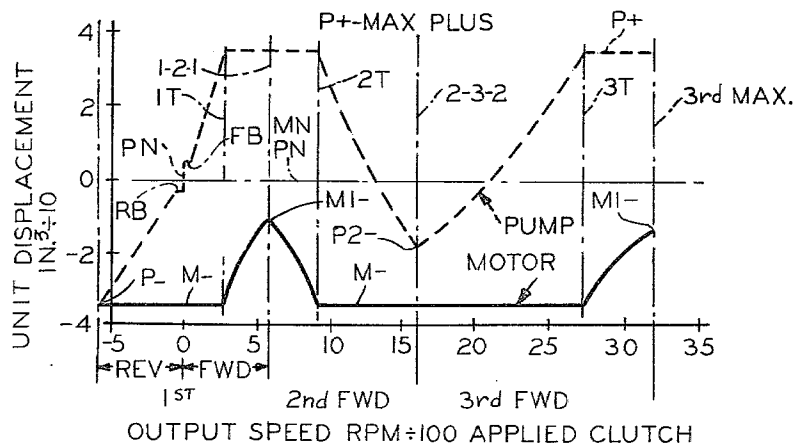

UNIT DISPLACEMENT IN.³÷10
P+-MAX PLUS
P+
1-2-1
1T
2T
2-3-2
3T
3rd MAX.
PN
FB
MN PN
RB
MI-
P2-
PUMP
MOTOR
P-
M-
REV
FWD
1ST
2nd FWD
3rd FWD
OUTPUT SPEED RPM÷100 APPLIED CLUTCH

Fig. 4

TRANSMISSION AND CONTROL SYSTEM

The invention hereindescribed was made in the course of work performed under a contract or subcontract thereunder with the United States Department of the Army.

FIELD OF INVENTION

This invention relates to transmissions and controls therefor and, particularly, cross drive hydromechanical transmissions and controls therefor.

RELATED APPLICATIONS

Commonly assigned U.S. Patent Applications Ser. NO. 051,109 now U.S. Pat. No. 4,258,548 in the names of Arthur Hall III and Warren E. Leet; and Ser. No. 051,110 now U.S. Pat. No. 4,244,279 in the names of Max E. Stewart and James R. Lucas, filed concurrently on June 22, 1979, are related applications.

BACKGROUND OF INVENTION

A related cross drive transmission and controls therefor are shown in U.S. Pat. No. 3,611,838 Robert C. Utter, granted Oct. 12, 1971. A related hydromechanical transmission is shown in U.S. Pat. No. 3,982,448 James C. Polak and Phillip J. Ross, granted Sept. 28, 1976. Related hydrostatic transmissions and units thereof are shown in U.S. Pat. Nos. 3,274,946 Edw. E. Simmons, granted Sept. 27, 1966 and 3,752,010 R. G. Tipping, granted Aug. 14, 1973. The type of modulator valve used is shown in U.S. Pat. No. 3,093,010 W. M. Spreitzer et al., granted June 11, 1963. Speed governors, as shown in U.S. Pat. No. 2,762,384 Rosenberger, granted Sept. 11, 1956, are used. A related ratio governor is shown in U.S. Pat. No. 3,583,154 Robt. C. Utter, granted June 8, 1971. Manual electric controls are shown in U.S. Pat. No. 3,640,157 Robt. H. Schaefer, granted Feb. 8, 1972. A related zero stroke bias valve is shown in U.S. Pat. No. 3,590,577 Robt. C. Utter, granted July 6, 1971.

SUMMARY OF INVENTION

The cross drive transmission has a hydromechanical propulsion drive and a hydrostatic steer drive, both being driven by an input drive and driving right and left differential output drives for straight propulsion drive and differential steer drive. The hydromechanical propulsion drive is a multispeed (preferably with three ratio ranges) hydromechanical, synchronous shifting transmission providing continuously variable ratio change. The steer drive is a continuously variable ratio transmission, such as a hydrostatic transmission. The input drive barrel and the output drive barrel, both of which displace a small volume of fluid, are located in the bottom of the housing. The hydromechanical propulsion transmission and hydrostatic steer transmission barrel is located above these input and output barrels so the sump in the bottom of the housing has a large fluid capacity volume at low fluid levels.

This control system is designed for use with a propulsion hydromechanical transmission in which the motor of the hydrostatic portion of the drive provides driving torque during one drive ratio and reaction torque during another drive ratio, and wherein its shifts from one ratio to another are designed to be synchronized. The hydraulic ratio of this transmission is controlled by varying the displacement of a variable displacement pump and motor. In synchronous shifting hydromechanical transmissions, the gearing is designed so that the engaging friction members of the oncoming friction device for establishing 2nd and higher ratio ranges have slightly different rotational speeds before and after the shift to accommodate the exchange of efficiencies in the hydrostatic drive by a small change of motor speed for improved synchronization.

The manual control, on a shift from neutral to pivot steer or forward and reverse drive positions, controls a disconnect valve to establish disconnect clutch drive to the propulsion and steer drives and to make the steer controls operational. The manual control, on a shift from neutral or pivot steer to forward or reverse drive positions, controls the neutral valve to also connect the advance signal so both the advance and return signals are connected to the propulsion hydrostatic control, and supplies the shift valves so both the propulsion hydrostatic and ratio shift controls are operational. Also, the forward and reverse valves, in forward and reverse manual positions, respectively provide forward and reverse signals. In forward and reverse drive, in 1st ratio, the return signal is connected to a return signal valve and a zero stroke bias valve to the actuator, and the advance signal is connected by the advance signal valve to the actuator for changing pump and motor displacement. In 2nd and 3rd ratio, the return signal is connected by the return signal valve to the advance signal control valve which supplies both return and advance signals to the actuator for changing both pump and motor displacement. The advance signal control valve reverses the connections in 2nd and reverse ratio range to provide reversed displacement change relative to input speed.

To accomplish the above shift requirement, a motoroperated shift signal valve has a spool operatively connected for movement with the displacement member of the hydrostatic motor to provide a 1-2-1 shift signal at the desired motor displacement for 1-2-1 shifting. A similar pump-operated shift signal valve provides a 2-3-2 shift signal at the desired pump displacement for 2-3-2 shifting. These 1-2-1 and 2-3-2 shift signals are connected by a ratio shift signal valve in upshift or downshift positions respectively in response to the advance or return signal to provide 1-2 and 2-3 upshifts and 2-1 and 3-2 downshift signals to control shift valves to effect such shifts. The shift valves lock in upshift position until a downshift signal effects a downshift.

A plurality of exhaust trip valves are associated with the shift valves, providing a restricted exhaust and restricted supply to maintain the off-going friction device pressurized until the oncoming friction device has sufficient engagement pressure to carry the torque and to discontinue the restricted supply.

A ratio governor is responsive to input speed, compares input speed to a varied desired set speed, and operates as a regulator to provide equal low pressure advance and return signals on a speed match, to quickly increase the advance signal to maximum value and reduce the return signal to zero on overspeed, and to quickly increase the return signal and reduce the advance signal to zero on underspeed, to control pump and motor displacement to maintain input speed at the set speed. These signals are distributed by the control system to an actuator which controls the movement of servo valves and hydraulic servos to control the displacement member of the hydraulic pump and motor. The ratio governor signals are supplied to the ratio shift

signal valve which is positioned with hysteresis in an upshift or a downshift position, depending on the signal emitted by the ratio governor, for controlling the ratio range shift valves.

In the ratio governor, the set speed is controlled by a plurality of vehicle, engine, transmission, and brake operation signals, and the rate of change of the advance and return signals is controlled for improved governor operating characteristics or stability. The set speed is varied from a low speed-such as engine idle speed, to a high or maximum engine speed, by control signals proportional to each of leading torque demand-such as the degree of throttle position; steer demand-such as the degree of steer lever movement from straight drive position; engine power or substantially simultaneous torque demand-such as compressor discharge pressure from a gas turbine engine; and vehicle brake demand-such as vehicle brake apply pressure. The set speed will be the highest set speed called for by any one of these control signals.

In drive operation during acceleration, increasing throttle, in a low-throttle range, will increase the set speed to an intermediate value. Then, as engine power increases to a high or maximum value, the set speed is increased to the maximum. As throttle is decreased for deceleration, the lagging decreasing engine power signal controls the set speed. Full throttle and maximum engine power may increase the set speed to the same maximum value. The steer demand increases the set speed to a higher value to maintain higher engine speed, even though throttle and engine power are low, to maintain a greater degree of steer capability. During braking in higher ratio ranges, the brake demand increases the set speed to a higher value for increased engine braking.

To avoid a "stumble" effect as the vehicle speed slows to zero, the vehicle brake apply signal is fed through a brake signal valve which blocks feed of the vehicle brake apply signal to the ratio governor in 1st ratio range. To ensure a smooth set speed transition to idle as 1st ratio range engages during a vehicle brake stop, the 1st range signal to the brake signal valve is fed through a restriction into an actuator chamber for accumulator operation delaying the transition. The vehicle brake apply signal thus has a modulated transition from maximum to zero signal value at a controlled rate to provide a smooth set speed reduction.

During normal operation, the advance and return signals are connected through an advance cut-off valve and a rate of change control to provide ratio governor system stability and smooth ratio change. To obtain these advantages, the return signal, which is provided in response to any request for a reduction in ratio, is supplied rapidly by full-flow lines to quickly provide full pressure for fast reduction of the ratio of the propulsion drive provided by the propulsion hydromechanical transmission. Also, for fast reduction of ratio, the advance line is rapidly exhausted by a full-flow line. When the advance signal is provided to increase ratio, it is supplied slowly by a restricted line for slowed increase of the ratio, and the return line is exhausted slowly by a restricted line.

Since the hydrostatic ratio change in 1st ratio range operation provides a larger percentage change in speed, the transmission is more sensitive to rate of change of ratio. Thus, to provide optimum ratio change stability and shift smoothness in lower ratio ranges such as 1st ratio range, the rate of increase in the advance signal is further decreased by including a second restriction in the advance line. This system provides an advance signal rate of change increasing with increasing gear speed ratio or ratio range.

The transmission has the capability to decrease the steering radius to the sharpest turn, pivot steer, and the steer hydrostatic transmission has a large power capacity. Since a steer request for a sharp turn at low engine speed, such as idle, may stall the engine, the steer request controls include a system to prevent stalling the engine at low input speed. The manual steer request linkage includes a steer override link which, when supplied with fluid pressure by a steer inhibitor or signal valve, transmits steer request to the steer servo valve. At low speeds, the steer signal valve cuts off fluid pressure supply to the steer override link so it does not transmit steer request. The fluid pressure supply to the steer signal valve is cut off by a regulator valve in response to abnormally low minimum charging pressure in the steer hydrostatic transmission to prevent steering. The manual control for neutral controls the steer signal valve to prevent steer.

The propulsion hydromechanical transmission has in the hydrostatic transmission portion, an overpressure protection system for limiting high hydrostatic pressure which includes a stroke limiter control operative in response to overpressure of high hydrostatic pressure to limit stroke to limit high hydrostatic pressure and a drive control operative in higher ratio ranges, 2nd and 3rd, in response to overpressure of high hydrostatic pressure, even though the stroke limiter control is operating, to an overpressure cut-off valve to cut off the mainline feed to all the ratio-establishing devices to place the transmission in neutral and to lock in this cut-off mode substantially until stroke is zero and output governor pressure and speed are zero and then unlocking so operation of the ratio controls may be resumed with normal 1st ratio range starting operation. Also in this cut-off mode, the overpressure cut-off valve provides a return demand signal. Also lower than normal charging or low kidney pressure in the propulsion hydrostatic transmission also provides a return demand signal. Both of these return demand signals control an advance cut-off valve of the ratio governor to cut off the advance signal and continuously provide a return signal.

These and other features and advantages of this invention will be more apparent from the following description and accompanying drawings.

BRIEF DESCRIPTION OF DRAWINGS

FIG. 1 is a gear schematic of the cross drive transmission on line 1—1 of FIG. 2.

FIG. 2 is a cross-sectional view, with portions broken away, of the cross drive transmission gearing.

FIG. 3 is a block diagram showing the relative positions of FIGS. 3A, 3B, 3C, 3D, 3E, and 3F.

FIGS. 3A, 3B, 3C, 3D, 3E, and 3F, when arranged as shown in FIG. 3, provide a diagrammatic view of the propulsion and steer controls for the cross drive transmission.

FIG. 4 is a plot of pump and motor displacement relative to output speed for the propulsion hydrostatic transmission.

DESCRIPTION OF PREFERRED EMBODIMENTS

GEARING

The transmission has a cross drive train **10** (FIGS. **1** and **2**) and a control system **11** (FIGS. **3A** to **3F**). Cross drive train **10** has an input drive **12** which is connected by both a propulsion drive **13** and a steer drive **14** to drive the right and left output drives **16** and **17**. In input drive **12**, the engine-driven input shaft **18** is drive-connected by a spur gear type 1st input gearset **19** and bevel gearset **20** to drive the cross input shaft **21**. The 1st input gearset **19**, in the preferred arrangement, provides a speed increase, so cross drive train **10** can be used with different engines. Bevel gearset **20** provides drive from the axially located input shaft **18** to the transverse or cross input shaft **21**.

The preferred control system **11** controls the propulsion drive **13** to provide forward and reverse drive. Also, in input drive **12**, cross input shaft **21** is drive-connected by a second input drive **22** having a disconnect clutch **24** connected to drive a 2nd input gearset **25** for driving final input shaft **26** which provides one of the two inputs for propulsion drive **13** and the input for steer drive **14**, as explained below. The 2nd input gearset **25** has an input gear **27** rotatably mounted on cross input shaft **21** and driven by disconnect clutch **24**, and an output gear **28** driving final input shaft **26**.

Right output drive **16** has a planetary output gearset **29** having an input ring gear **30** mounted on an input drum **31** which is driven by propulsion drive **13**, and an input sun gear **32** which is driven by steer drive **14**. Input ring gear **30** and sun gear **32** mesh with planetary pinions **33** rotatably mounted on carrier **34** having a drum portion **35** and being connected to right output shaft **36**. The right output brake **37**, preferably a fluid-operated and cooled multiplate brake, is located between carrier drum portion **35** and housing **38**. The planetary output gearset **29** has a propulsion drive to input ring gear **30** and a steer drive to input sun gear **32** and provides a differential drive to right output shaft **36**.

The left output drive **17** is a mirror image of right output drive **16** and functions in the same way, so like reference numerals (primed) have been used (FIG. **1**), and reference is made to the above description.

Propulsion drive **13** is a synchronous shifting hydromechanical transmission functioning to provide a wide range of continuously variable torque and speed ratio drives extending through three gear ratios, and is similar to FIG. **3** of the above U.S. Pat. No. 3,982,448. Propulsion drive **13** has a propulsion hydrostatic transmission **39** having a pump or H1 and a motor or H2 hydrostatic units **41** and **42**, and 1st, 2nd, and 3rd planetary gearsets **43**, **44**, and **45**, and 1st and 2nd output gearsets **47** and **48** of the spur gear type. The H1 and H2 hydrostatic units are known pump-motor units as shown, for example, in the above U.S. Pat. Nos. 3,274,946 and 3,752,010. The pump and motor hydrostatic units **41** and **42** are also called "H1" and "H2" hydrostatic units because at times H1 acts as a pump and H2 acts as a motor, and at other times H2 acts as a pump and H1 acts as a motor, depending on the direction of power transfer, as pointed out in the description of the operation.

The 1st planetary gearset **43** has planetary pinions **51** rotatably mounted on the carrier **52** and meshing with ring gear **53** and sun gear **54**. Final input shaft **26** is drive-connected to ring gear **53**, carrier **52** is drive-connected to H1 rotor shaft **56**, and sun gear **54** is drive-connected by intermediate shaft **57** to H2 rotor shaft **58**. Final input shaft is differentially drive-connected by 1st planetary gearset **43** to both the H1 and H2 rotor shafts **56** and **58**.

The 2nd planetary gearset **44** has planetary pinions **61** rotatably mounted on the carrier **62** and meshing with ring gear **63** and sun gear **64**. To establish 1st ratio range drive, the fluid-operated friction-type 1st brake **65** is engaged to hold carrier **62** to establish a speed reduction drive from intermediate shaft **57** which is drive-connected to sun gear **64** and provides a reduction gear drive from ring gear **63** to ring gear shaft **66**.

The 3rd planetary gearset **45** has planetary pinions **67** rotatably mounted on a carrier **68** and meshing with ring gear **69** which is drive-connected to ring gear shaft **66**, and sun gear **71** which is drive-connected to intermediate shaft **57**. To establish 3rd ratio range drive, the fluid-operated friction-type 3rd clutch **72** is engaged to provide a 3rd ratio drive **73** from cross input shaft **21** through 3rd clutch **72** and 3rd input gearset **76** having an input gear **77** on a clutch shaft **74** and an output gear **78** driving carrier **68**.

The 1st output gearset **47** has an input gear **81** on ring gear shaft **66**, and an output gear **82** on cross drive shaft **83** which is connected to drive the input ring gears **30** and **30'**, respectively, of right and left output drives **16** and **17** for 1st and 3rd ratio drives. The 2nd output gearset **48** has an input gear **86** fixed on H1 rotor shaft **56** and driving output gear **87** which is connected by 2nd clutch **88** (when engaged) to drive cross drive shaft **83** for 2nd ratio drive. The 2nd clutch **88** is a known fluid-operated multiplate friction clutch.

Steer drive **14** has a steer hydrostatic transmission **89** having pump (P) **90** driven by final input shaft **26**, and motor (M) **91** driving 1st steer drive shaft **92**. In steer drive transmission **89**, the hydrostatic units are designated pump (P) and motor (M) since during steer drive, the power transfer is from final input shaft **26** to pump (P) and from motor (M) to 1st steer drive shaft **92**. The steer drive gearset **93** has an input gear **94** on 1st steer drive shaft **92** connected by idler gear **94'** for driving output gear **95** and 2nd steer drive shaft **96**. The 2nd steer drive shaft **96** is connected to drive cross steer shaft **97** by a steer ratio pack **98**—which may be a direct connection providing only one ratio drive, or may be a known 3-speed transmission. When steer ratio pack **98** is a 3-speed transmission, the vehicle operator may manually select 1st speed for pivot steer, 2nd speed for normal steer, and 3rd speed for water steer by track drive of amphibious vehicles. The 3-speed transmission may be a manual or powershift type transmission having countershaft or planetary gearing, or a hydromechanical type transmission combining 3-range gearing with steer hydrostatic transmission **89**, such as the above propulsion drive **13**.

The cross steer shaft **97** and right steer drive gearset **101** and left steer drive gearset **102** provide a differential steer drive **103** respectively to right and left input sun gears **32** and **32'** of right and left output drives **16** and **17**. Right steer drive gearset **101** has an input gear **104** driven by cross steer shaft **97** and an output gear **105** driving right input sun gear **32**. Left steer drive gearset **102** has an input gear **106** driven by cross steer shaft **97**, an idler gear **107**, and an output gear **108** driving left input sun gear **32'**. Thus, right and left input sun gears

32 and 32' are driven in opposite directions at the same speed relative to cross steer drive shaft **97**.

Cross drive train **10** (FIGS. **1** and **2**) is arranged in barrels, so sump **111** has a high fluid volume capacity in the lower portion of housing **38**. Each barrel has a shaft and drive elements concentric therewith. Input barrel IB has cross input shaft **21** and the drive elements concentric therewith, a bevel gear of bevel gearset **20**, gears **27** and **77**, 3rd clutch **72**, and disconnect clutch **24**. Output barrel OB has cross drive shaft **83** and the drive elements concentric therewith, right and left output drives **16** and **17**, gears **82** and **87**, and 2nd clutch **88**. Transmission barrel TB has final input shaft **26** and intermediate shaft **57** and the drive elements concentric therewith, 1st, 2nd, 3rd planetary gearsets **43**, **44**, and **45**; gears **28** and **78**, and propulsion and steer hydrostatic transmissions **39** and **89**. Differential drive barrel DB has cross steer shaft **97** and the drive elements concentric therewith, gears **95**, **104**, and **106**, and steer ratio pack **98**. Input barrel IB and output barrel OB are located in the lower portion of housing **38** which closely surrounds the lower portions of barrels IB and OB. Barrels IB and OB have drive elements displacing a small volume of fluid (as compared to that of transmission barrel TB), so sump **111** has a large fluid volume, up to fluid level L (FIG. 2).

Transmission barrel TB, which has a large fluid volume because it has both the propulsion and steer transmissions, is between and above input barrel IB and output barrel OB and the fluid level L. Differential barrel DB is above output barrel OB. This arrangement is very compact so an essentially square housing **38** closely encloses cross drive train **10**. The rotary components are supported by bearings supported on housing **38**, and fixed components are fixed to housing **38**.

CONTROL SYSTEM

The control system **11** (FIGS. **3A** to **3F**) has portions similar to the above U.S. Pat. No. 3,611,838, and controls a cross drive transmission having a propulsion and a steer drive **13** and **14**, having hydrostatic transmissions **39** and **89**.

Referring to FIG. **3F**, a main pump **110** supplies fluid from sump **111** to mainline **112** extending through a main relief valve **113** which exhausts main pressure fluid back to sump **111** when above regulated mainline pressure.

Mainline **112** supplies control system **11** at mainline regulated fluid pressure regulated by a known main regulator valve **114** which supplies overage fluid to lube line **115**. Mainline **112** distributes main pressure fluid to the basic signal controls **116** (FIG. **3C**) for control system **11**, including modulator valve **117**, output speed governor **118**, governor signal valve **119**, and input speed governor **120**. Mainline **112** also supplies, in the propulsion controls **121**, advance cut-off valve **122** which is associated with 1st range advance valve **123**, return demand valve **124**, disconnect clutch valve **125** (FIG. **3B**), inhibitor valve **126**, neutral shift valve **127**, and forward signal valve **128** which controls the supply to reverse signal valve **129**. Mainline **112** also supplies hydrostatic drive controls **131** (FIGS. **3A** and **3D**), zero stroke indicator valve **132**, propulsion and steer hydrostatic charging systems **134** and **135**, hydrostatic control pump **136**, and low charge signal valve **631**. Mainline **112** is also the exhaust line for propulsion and steer control pressure regulator valves **138** and **139**. Mainline **112** is connected by check valve **137** to propulsion control pressure line **564**. Mainline **112** also supplies, in the vehicle service brake system **141** (FIG. **3E**), brake apply regulator valve **142**.

MODULATOR VALVE

The modulator valve **117**, shown in FIG. **3C**, is a valve of the type shown in the above U.S. Pat. No. 3,093,010, and is supplied with mainline fluid pressure by mainline **112** and controlled by compressor discharge pressure from a gas turbine engine supplied by compressor discharge pressure line **144**, to provide an engine power pressure signal, which increases in proportion to increasing compressor discharge pressure, in engine power signal line **146**. Since compressor discharge pressure provides a torque demand signal substantially simultaneous with engine power, this type modulator valve controlled by compressor discharge pressure may be used to provide an engine power pressure signal.

OUTPUT SPEED GOVERNOR

The known output speed governor **118** (FIG. **3C**) is driven by a cross drive shaft **83**, supplied by mainline **112**, and provides output or 2nd governor pressure increasing with output speed in output or 2nd governor pressure line **147**. A known two-weight governor providing a stepped governor pressure curve approximating a straight line, with governor pressure increasing with speed, e.g., as shown in the above U.S. Pat. No. 2,762,384, is preferred.

OUTPUT GOVERNOR SIGNAL VALVE

The output governor signal valve **119** has a valve spool **151** having two equal-diameter lands a and b slidable in a valve bore **152**. Valve bore **152** and the other valve bores are in a known valve body **169** fixed in housing **38** and having several parts to conveniently locate the valves-except for the bores described as being in movable sleeves.

Spring **153** is located in spring chamber **154**, which is an end portion of valve bore **152** vented by exhaust **156**, and is seated on the end of bore **152** and engages land **151*b*** to bias valve spool **151** to the exhaust position shown (FIG. **3C**), blocking mainline **112** and connecting output or 2nd governor signal pressure line **157** to exhaust **158**. Output or 2nd governor pressure line **147** is connected to governor pressure chamber **159**, an opposite closed end portion of bore **152**, and acts on land **151*a*** to bias valve spool **151** to a pressure-increasing position, blocking exhaust **158** and connecting mainline **112** to 2nd governor signal pressure line **157**. Thus, governor signal valve **119** regulates 2nd governor signal pressure to increase from zero pressure at an intermediate output governor pressure value or output speed, and to increase with output speed at the same rate as output governor pressure.

INPUT SPEED GOVERNOR ASSEMBLY

The input speed governor assembly **120** (FIG. **3C**) is, in part, similar to governors (**114**, **116**) in the above U.S. Pat. No. 3,611,838. Input speed governor assembly **120** has an engine governor **161** and a ratio governor **162** which are driven by governor shaft **163**, and a governor gear **164** driven by the engine or input shaft **18**. The known engine governor **161** is like the governors shown in the above U.S. Pat. Nos. 2,762,384 and 3,611,838. This two-weight type engine governor **161** is driven at engine speed, supplied by mainline **112**, and provides an

input or 1st governor pressure increasing with increasing engine speed in 1st or input governor pressure line **165** which is connected to the steer signal valve **166** (FIG. 3D).

ADVANCE RETURN SIGNAL SYSTEM

The advance return signal system **160** is a governing system and has ratio governor **162** of input speed governor assembly **120**; advance cut-off valve **122**; 1st range advance control valve **123**; return demand valve **124**; restriction and check valve device **229**; and rate of ratio change control **203** having restriction and check valve devices **233** and **249**, for supplying advance or return signals to advance or return line **204** or **205**.

RATIO GOVERNOR

The ratio governor **162**, as described below, has some portions similar to the ratio governors shown in the above U.S. Pat. No. 3,611,838, and new portions.

The ratio governor **162** includes a ratio governor valve **170** having a valve spool **167** having four equal-diameter, spaced lands a, b, c, and d, slidably disposed in a valve bore **168** in valve body **169**. A pair of flyweights **171** of the flyweight assembly **172** are rotated by governor shaft **163** so that an outward force proportional to engine speed is imposed on the right end of valve spool **167**. A spring **173** is compressed between a spring seat **174** fixed on the other end of valve spool **167** and the end wall **176** of piston **177** slidable in a bore **178** of control cylinder **179** having an end wall **180**. A chamber **181** in control cylinder **179**, between end wall **180** and end wall **176** of piston **177**, is continuously connected by transfer port **182** to power signal line **183** which supplies a pressure proportional to engine power or brake apply pressure. The higher pressure in engine power signal line **146** or vehicle brake apply signal line **184** is connected by power signal shuttle valve **186** (a known shuttle valve) to power signal line **183**. Control cylinder **179** is slidably mounted in a bore **187** in valve body **169**. There is sufficient clearance for full control movement of both piston **177** and control cylinder **179**. Either or both throttle cam **188** or steer request cam **189** abuts the end wall **180** of control cylinder **179** to variably position control cylinder **179**. Throttle and steer request cams **188** and **189** are respectively fixed on throttle and steer request shafts **191** and **192** which are concentrically rotatably mounted on a support portion **193** of transmission housing **38**. The engine throttle **194** is connected by throttle linkage **195** to rotate throttle shaft **191** and cam **188** in proportion to throttle position or leading torque demand. The steer request lever **196**, which is controlled by the driver to request steer, is connected by steer request linkage **197** to rotate steer request shaft **192** and cam **189** in accordance with steer request.

In this throttle and steer request device **190**, the throttle cam **188** or steer request cam **189** moves control cylinder **179** which engages and moves piston **177**—or power pressure in chamber **181** moves piston **177**—so that spring **173** provides a bias force on valve spool **167** which is equal to the highest of the bias forces provided in response to the control functions, throttle demand, steer request, engine power, or brake apply pressure. This bias force determines the set speed, a predetermined variable speed value in a range from engine idle speed to maximum engine speed, at which ratio governor **162** regulates engine speed by controlling the propulsion drive ratio.

In ratio governor valve **170**, the central lands b and c of spool **167** block bore **168** and have a small land overlap relative to the ports for advance and return feed lines **201** and **202**, while end lands a and d of spool **167** have a through-passage or groove for exhaust. Mainline **112** is connected to bore **168** between lands b and c of spool **167**. When engine speed equals or matches set speed, mainline pressure is connected through the land overlap to both advance and return feed lines **201** and **202**; and both lines are connected through the overlap respectively through lands a and d of spool **167** to exhaust, so both the advance and return signal pressures have low values. On engine overspeed (engine speed greater than the set speed), mainline pressure is supplied to advance feed line **201** to increase advance signal pressure to mainline pressure, and return feed line **202** is exhausted through land d of spool **167**. On engine underspeed (engine speed less than the set speed), mainline pressure is supplied to return feed line **202** to increase return signal pressure to mainline pressure, and advance feed line **201** is exhausted through land a of spool **167**. An increased degree of movement of valve spool **167** from a central position in the range of the land overlap increases the rate of pressure change of the advance and return signals.

On movement from its minimum (idle) to full (maximum) throttle position, engine throttle **194** rotates throttle cam **188** to act on and move control cylinder **179** and piston **177**, and spring seat **174** to increase the bias force of spring **173** and to increase the set speed of ratio governor **162** from minimum (idle) to maximum set speed—such as full power engine speed or maximum engine speed.

On movement from its minimum to maximum positions, the null or straight steer position, to either right or left full or maximum steer request positions, the steer request lever **196** rotates steer request cam **189** in either direction. Similar right and left portions of steer request cam **189** act similarly to increase the bias force of spring **173** to increase the set speed from a minimum (idle) to a high set speed, e.g., 80% of maximum engine speed.

As set forth above and further explained below, each control functions at its minimum value—idle throttle demand, zero steer request, idle engine power or compressor discharge pressure, and brake disengaged or zero brake apply pressure operates to control ratio governor **162** to provide the minimum, or engine idle set speed. Then, as the control functions are increased, the set speed is increased from idle speed to higher engine speeds. The set speed will be the highest set speed called for by any one of these control functions which individually operate independently or, in the absence of any other control function, as follows.

The power pressure signal in power signal line **183** is alternatively the engine power signal pressure or vehicle brake apply pressure, and acts in control cylinder chamber **181**. Since control cylinder **179** abuts at least one of the throttle and steer request cams **188** and **189**, there is a reaction; and when the power pressure has a sufficiently high value to provide a higher bias force on spring **173** than that provided by throttle and steer request cams **188** and **189**, the power pressure moves piston **177** to increase the bias force of spring **173** and the set speed from a minimum to a maximum set speed. The minimum set speed for each of these control functions, torque demand, steer request, engine power and brake torque at their minimum value, is engine idle speed. Control cylinder **179** will be positioned by the

one of throttle and steer request cams **188**, **189**, calling for the higher set speed, and power pressure in chamber **181** will not move piston **177** away from abutting relation with control cylinder **179** to increase the bias force of spring **173**, unless the power pressure calls for a higher set speed. Thus, the set speed will be the highest set speed called for by one of these control functions—but not higher because these control functions forces on spring **173** are not added.

The throttle control, on movement from its minimum to maximum position, will increase the set speed from idle speed to a high or maximum set speed, e.g., full power or maximum engine speed. Then, after a time delay as engine power increases to a maximum, the engine power signal supplied by compressor discharge pressure will increase the set speed to the maximum engine speed. This arrangement provides ratio change in response to torque demand shortly before the resultant change of engine torque and power during engine acceleration, and ratio change in response to actual engine torque and power during deceleration.

The steer request cam **189**, as steer request varies in the full range (zero to maximum), controls the set speed in a wide speed range (e.g., idle to 80% of maximum speed), to maintain engine speed at higher values relative to propulsion output speed to maximize steer operation—even though the operator decreases throttle demand. At higher engine speeds, steer drive **14**, with the same steer ratio will provide a greater speed difference between the right and left output drives **16** and **17**, and thus the output shafts **36** and **36'** have a larger differential speed for tighter or smaller radius turns. When steering, the vehicle operator often reduces the throttle to reduce propulsion power, but this steer request control prevents a fully proportional reduction of steer capability and maintains a higher proportion of maximum steer capability.

Normally when the vehicle service brake system **141** is applied for vehicle braking, the throttle **194** is returned to Idle and the power pressure signal, based on compressor discharge pressure, will then return to Idle. Thus, during braking—as the engine power pressure signal decreases and vehicle brake apply pressure increases—the power signal shuttle valve **186** connects the higher of these pressures via power signal line **183** to chamber **181** of ratio governor **162** to maintain the set speed and thus engine speed at higher values, thus maintaining higher engine braking and higher steer capability during vehicle braking.

RETURN DEMAND CONTROL

Return demand control **206** employs an advance cut-off valve **122** controlled by return demand valve **124** which is controlled by a return demand signal supplied by return demand signal line **227** which, in turn, is supplied by **2-3** overpressure cut-off valve **431** or low charge signal valve **631**, as explained below.

ADVANCE CUT-OFF VALVE

The advance cut-off valve **122** has a spool **208** having three equal-diameter lands a, b, and c, slidable in a bore **209**. Operating pressure, which normally is supplied by operating line **211** to closed operating chamber **212** at one end of bore **209**, moves spool **208** against spring **213** to engage stop **215** in the connecting position shown in FIG. **3C**. Spring **213** is located in spring chamber **214** which is located at the opposite end of bore **209** and is vented by an exhaust **216**. In this normal connecting position of advance cut-off valve **122**, advance and return feed lines **201** and **202** are respectively connected, between lands a and b and lands b and c of spool **208**, to advance and return connecting lines **217** and **218**. When operating line **211** is exhausted, spring **213** moves spool **208** to the cut-off position so advance cut-off valve **122** functions so that lands a and b of spool **208** respectively block advance and return feed lines **201** and **202**, advance connecting lines **217** is connected between lands a and b of spool **208** to exhaust **219**, and mainline **112** is connected between lands b and c of spool **208** to return connecting line **218**.

RETURN DEMAND VALVE

Return demand valve **124** has a spool **221** having two equal-diameter lands a and b slidable in a bore **222**. Spool **221** is biased by spring **223** to the normal connecting position shown (FIG. **3C**), blocking exhaust **226** and connecting mainline **112** to operating line **211**. Spring **223** is located in one end of bore **222** which is vented by exhaust **224**. The return demand signal line **227** is connected to chamber **228** to move spool **221** against spring **223** to the cut-off position, blocking mainline **112** and connecting operating line **211** to exhaust **226** permitting advance cut-off valve **122** to be moved by its spring **213** to the advance cut-off position. A timing or restriction and check valve device **229** having restriction **231** and check valve **232** is placed in operating line **211** for delayed flow only through the restriction **231** to operating chamber **212** to delay movement of advance cut-off valve **122** to the normal connecting position and rapid exhaust flow from operating chamber **212** through both the restriction **231** and check valve **232** for rapid movement of advance cut-off valve **122** to the cut-off position for rapid cut-off of advance pressure and supply of return pressure and delayed return to normal position for connection of advance and return pressures as supplied by ratio governor **162**.

RATE OF RATIO CHANGE CONTROLS

To provide, in the advance return signal system **160**, governor stability and smooth ratio change, rate of ratio change controls **203**, between the advance and return feed lines **201** and **202** and advance and return lines **204** and **205**, provide a high rate of flow and pressure increase of return pressure to return line **205** for rapid response to a request for reduction of drive ratio and a lower rate of flow and pressure increase of advance pressure to advance line **204** for slower response to a request for an increase of drive ratio.

The advance connecting line **217** is connected by a restriction and check valve device **233** and 1st range advance control valve **123** to advance line **204**. The 1st range advance control valve **123** has a spool **234** having equal-diameter lands a and b slidable in a bore **236**. Spring **237**, in one end of bore **236** vented by exhaust **238**, biases spool **234** to the higher range position shown in FIG. **3C**. The 1st apply pressure, which is supplied by 1st apply line **239** to operating chamber **241** located at the opposite end of bore **236**, moves spool **234** against spring **237** to the low or 1st range position. When advance pressure is supplied from advance connecting line **217** to restriction and check valve device **233**, the check valved line **242** is closed and blocks flow, and flow is only via restricted line **243** to 1st range advance control valve **123** which, in higher 2nd and 3rd range positions, connects restricted line **243** via its free-flow advance branchline **244** to advance line **204**, and in 1st range

position connects restricted line 243 via its restricted branchline 245 to advance line 204. Thus, in 1st range the supply of advance pressure flow is restricted by two restrictions in series—restricted line 243 and restricted branchline 245. So the delay in advance pressure flow and pressure increase in 1st ratio range is greater than in higher ranges wherein advance flow is only through one restriction, restricted line 243. When advance pressure decreases, the flow is from advance line 204 to the now open check valved line 242 which by-passes 1st range advance control valve 123 for flow to advance connecting line 217 to exhaust at advance cut-off valve 122 or ratio governor 162.

The return connecting line 218 is connected through both check valve 247 and restriction 248 of check valve and restriction device 249 for fast flow to and rapid increase of return pressure in return pressure line 205. When return pressure decreases, the reverse direction flow is only through restriction 248 and via advance cut-off valve 122 for exhaust at ratio governor 162 for slow decrease of return pressure.

In the rate of ratio change controls 203, normally there is no return demand signal pressure in return demand signal line 227, so the return demand valve 124 supplies operating pressure via operating line 211 to position advance cut-off valve 122 in the connecting position shown (FIG. 3C) for connecting advance and return feed lines 201 and 202 respectively to advance and return connecting lines 217 and 218. Return feed line 202 and connecting line 218 are connected via both open check valve 247 and restriction 248 for rapid supply to return line 205, and advance line 204 is connected by both open check valved line 242 and restricted line 243 for rapid exhaust to advance connecting line 217 and feed line 201 for rapid response to a request for a reduction in the ratio of the propulsion drive. Conversely, advance pressure is supplied with delay from advance connecting line 217 only through one restriction—restricted line 243, to advance line 204 for delay in the higher ratio drives (2nd and 3rd), and through two restrictions—restricted line 243 and restricted advance branchline 245, for greater delay in low ratio drive (1st); and return line 205 is exhausted only via restriction 248 for delay of exhaust of return line 205, so the propulsion hydrostatic transmission 39 and propulsion drive 13 have a delayed response to a request for increase in drive ratio.

In 1st range which uses 1st gear ratio, the vehicle is accelerated from zero speed, so ratio changes represent large percentage changes in speed. The drive system is therefore more sensitive to rate of ratio change in 1st range than it is in higher transmission gear ratios. To further desensitize ratio change in propulsion drive 13 in 1st range, 1st range clutch pressure operates on 1st range advance control valve 123 (as explained above) and provides a 2nd restriction, restricted advance branchline 245 in series with a 1st restriction, restricted line 243, in the advance pressure feed to advance line 204. Therefore, rate of ratio change controls 203 provide a variable advance signal pressure rate of change based on the transmission operating ratio range.

The return demand signal in return demand signal line 227 operates return demand valve 124 to exhaust operating line 211 quickly through full-flow check valve 232 to quickly move advance cut-off valve 122 for rapid supply of return pressure from mainline 112 to return line 205 and rapid exhaust of advance pressure from advance line 204. The restriction and check valve devices 233 and 249, as pointed out above, provide rapid supply of return pressure and exhaust of advance pressure. Mainline 112 is used to supply return pressure, so this supply of return pressure is continuous and not dependent on the operation of ratio governor 162. The return demand signal line 227 is supplied by return demand shuttle valve 251 (FIG. 3F) when a fluid pressure signal is supplied by either or both overpressure signal line or low charging pressure signal line 253, respectively, indicating excessive high control or kidney pressure and less than normal charging or low kidney pressure in propulsion hydrostatic transmission 39, as explained below.

PROPULSION RATIO CONTROLS

The advance line 204 is connected to neutral shift valve 127 (FIG. 3B). The return line 205 is connected to neutral shuttle valve 256 (FIG. 3B), and to ratio shift signal valve 257 (FIG. 3E). These and the other components of propulsion ratio control 121 are now described.

MANUAL RANGE CONTROL

A known manual control 254 (FIG. 3B), having an electric supply, manually controlled switches, and an electric circuit having a connection a to h to each grounded solenoid valve A to H, as shown e.g., in the above U.S. Pat. No. 3,640,157, energizes the solenoid valves A to H in each manual range position, as indicated by "X" in the following TABLE I.

TABLE I

MANUAL RANGE		SOLENOID VALVES A	B	C	D	E	F	G	H
FORWARD	(F)		X	X		X			X
NEUTRAL	(N)	X		X			X	X	
PIVOT STEER	(PS)	X		X			X		X
REVERSE	(R)		X		X		X		X

Each known solenoid valve (A to H) is connected to an associated valve operating pressure chamber, as explained below. There is a restricted feed to each operating chamber. The associated solenoid valve, when deenergized, is normally closed to provide pressure in the associated chamber, and when energized, opens to exhaust the associated chamber.

The manual range control 254 controls hydraulic range controls 255 which includes neutral shift valve 127, inhibitor valve 126, disconnect clutch valve 125, and forward and reverse signal valves 128 and 129.

NEUTRAL SHIFT VALVE

The neutral shift valve 127 (FIG. 3B) has a spool 258 having five equal-diameter lands a, b, c, d, and e, slidably disposed in bore 259. A compression spring 261, in 1st chamber 262 at one end of bore 259, engages land a of spool 258 and urges spool 258 to the right so that land e of spool 258 abuts the other end of bore 259 which encloses 2nd chamber 263. 1st and 2nd chambers 262 and 263 are closed. The 1st chamber 262 adjacent land a of spool 258 is connected to the solenoid valve B, and the 2nd chamber 263 adjacent the right end of land e of spool 258 is connected to solenoid valve A. Bore 259 is connected to mainline 112, a neutral advance line 264, automatic advance line 266, advance line 204, range feed line 267, 1st and 2nd chamber feed lines 268 and 269, and exhausts 273, 274, and 276. The 1st chamber feed line 268 is connected through restriction 271 therein to solenoid valve B for pressure control and to

supply mainline fluid pressure to, or exhaust from, 1st chamber **262** and indicator valve **126**, respectively, when neutral shift valve **127** is in the neutral position or drive position. Solenoid valve B retains or exhausts this pressure downstream of restriction **271** in 1st chamber **262** and supplied to inhibitor valve **126** respectively when solenoid valve B is deenergized or energized. Mainline **112** is always connected through restriction **270** to 2nd chamber feed line **269** which is connected to solenoid valve A for pressure control, and to supply pressure to, or exhaust from, 2nd chamber **263** and inhibitor valve **126**, respectively, when solenoid valve A is deenergized or energized.

In Neutral and Pivot Steer Manual Ranges, the solenoid valve A is energized, so 2nd chamber **263** adjacent land e of spool **258** will be exhausted, and spring **261** and pressure in 1st chamber **262**—because solenoid valve B is deenergized—positions neutral shift valve **127** in the neutral position shown (FIG. 3B). In forward and reverse automatic drive manual ranges, solenoid valve B is energized to exhaust 1st chamber **262** adjacent land a of spool **258** and solenoid valve A is deenergized so pressure in 2nd chamber **263** acts on land e of spool **258** to cause the spool **258** to shift against spring **261** to drive position. Solenoid valve B and 1st chamber **262** are used so that if solenoid valve A is not energized in Neutral or Pivot Steer manual ranges, neutral shift valve **127**, in neutral position nevertheless will remain in neutral position.

Neutral shift valve **127**, in neutral position, connects mainline **112** to 1st chamber feed line **268** and through restriction **271** therein for control by solenoid valve B and to 1st chamber **262** and inhibitor valve **126**, advance line **204** to neutral advance line **264** for neutral operation of propulsion hydrostatic transmission **39**, range feed line **267** to exhaust **273** so no range ratio is established, and automatic advance line **266** to exhaust **274** so there is no automatic advance of ratio of propulsion hydrostatic transmission **39**. Neutral shift valve **127**, in drive position, connects advance line **204** to automatic advance line **266** for automatic advance of propulsion hydrostatic transmission **39**, and connects neutral advance line **264** to exhaust **276**. Also, in drive position of neutral shift valve **127**, mainline **112** is connected to range feed line **267** for range ratio shifting, 1st chamber feed line **268** is disconnected from mainline **112** and connected to exhaust **274**, so 1st chamber **262** is exhausted. Solenoid valve B is also energized for exhaust for this automatic forward-reverse drive position.

INHIBITOR VALVE

The inhibitor valve **126** (FIG. 3B), for inhibiting ratio change, has a spool **277** having four equal-diameter lands a, b, c, and d, slidable in bore **278**. Spool **277** is biased to the normal or 1st ratio range position shown by spring **279** in spring chamber **281** at one end portion of bore **278** vented by exhaust **282** when 2–3 feed line **283** is exhausted in 1st ratio range. Spool **277** is moved to 2–3 ratio range position in 2nd and 3rd ratio ranges when 2–3 feed line **283** supplies 2–3 feed pressure to 3rd chamber **284** at the opposite end of bore **278** to move spool **277** against spring **279**. Mainline **112**, 1st (B) and and 2nd (A) chamber feed lines **268** and **269**, 3rd (C and E) chamber feed line **285**, and exhausts **286** and **287** are connected to bore **278** for control by spool **277**. The letters within parentheses indicate the solenoid valves supplied by these chamber feed lines. When inhibitor valves **126** and its spool **277** are in 1st ratio range position shown (FIG. 3B), mainline **112** is connected only by restriction **270** to 2nd chamber feed line **269** and is connected between lands c and d of spool **277** to 3rd chamber feed line **285**, and 1st chamber feed line **268** is blocked from exhaust **286** to be operative and exhaust **287** is blocked. When inhibitor valve **126** and its spool **277** are in the 2–3 ratio range position, mainline **112** is connected between lands b and c of spool **277** for full-flow to 2nd chamber feed line **269** in addition to the connection by restriction **270**, and 1st and 3rd chamber feed lines **268** and **285** are respectively connected to exhausts **286** and **287** to exhaust 1st chamber **262** of neutral shift valve **127** and reverse chambers **296** and **296'** of forward and reverse signal valves **128** and **129**.

FORWARD AND REVERSE SIGNAL VALVES

The forward and reverse signal valves **128** and **129** (FIG. 3B) are supplied by mainline **112** and controlled by selectively energized solenoid valves C, D, E, and F, to selectively supply forward or reverse signal lines **288** or **289** when the manual range control **254** is, respectively, in forward or reverse range position. Both forward and reverse signal lines **288** and **289** are exhausted in neutral and pivot steer positions of the manual range control **254**.

The forward signal valve **128** has a spool **291** having three equal-diameter lands a, b, and c, slidable in a bore **292**. Spring **293**, located in forward pressure chamber **294** and forward shift pressure in forward pressure chamber **294** both act on land a of spool **291** to bias spool **291** to the forward position shown, even if there is reverse shift pressure in reverse pressure chamber **296**. Reverse shift pressure in reverse pressure chamber **296** at the opposite end of spool **291** acts on land c of spool **291** in the opposite direction to bias spool **291** to reverse position against spring **293** when forward pressure chamber **294** is exhausted.

Reverse signal valve **129** (FIG. 3B) has the same structure as forward signal valve **128** so the same reference numerals (primed) have been used for the structure with reference to the above description of the forward signal valve **128**, and the different connections and solenoid valves are now described.

In the forward signal valve **128**, bore **292** is connected to forward exhaust **297**, forward chamber feed line **298** and forward feed line **299**, mainline **112**, reverse feed line **301** and reverse exhaust **302**. Forward chamber feed line **298** has a restriction **303** and is connected to feed forward pressure chamber **294** and the pressure therein is controlled by solenoid valve D. The 3rd chamber feed line **285** is connected by reverse chamber feed line **304** which has restriction **306** to feed reverse pressure chamber **296**, and the pressure therein is controlled by solenoid valve C.

In the reverse signal valve **129**, bore **292'** is connected to reverse feed line **301**, reverse signal line **289**, exhaust **307**, forward signal line **288** and forward feed line **299**. Forward chamber feed line **308** having restriction **309** connects forward signal line **288** to feed forward chamber **294'**, and the pressure therein is controlled by solenoid valve F. The 3rd chamber feed line **285** is also connected by reverse chamber feed line **311** having restriction **312** to feed reverse pressure chamber **296'**, and the pressure therein is controlled by solenoid valve E.

When gear ratio control **420** is providing neutral or 1st ratio range for forward or reverse drive, the 2–3 feed line **283** is exhausted, so inhibitor valve **126** supplies 3rd

chamber feed line **285**, and forward and reverse signal valves **128** and **129** are moved to all manual range positions in response to manual range control **254**, as described below.

When the manual range control **254** is in forward position, in forward and reverse signal valves **128** and **129**, respectively, solenoid valves C and F are energized to exhaust reverse pressure chambers **296** and **296'**, so springs **293** and **293'** move both forward and reverse signal valves **128** and **129** to the forward position shown (FIG. 3B). When forward signal valve **128** is in forward position, forward exhaust **297** is blocked, mainline **112** is connected to feed forward chamber feed line **298** and forward feed line **299** and exhaust reverse feed line **301** to reverse exhaust **302**. When reverse signal valve **129** is in forward position, forward feed line **299** is connected to supply forward signal pressure to forward signal line **288**, reverse signal line **289** is connected to exhaust **307**, and reverse feed line **301** is blocked. Since mainline pressure is supplied to both forward chamber feed lines **298** and **308** which respectively feed forward pressure chambers **294** and **294'**, solenoid valves D and F are deenergized and thus closed, so both forward pressure chambers **294** and **294'** are pressurized to hold both forward and reverse signal valves **128** and **129** are in forward position supplying forward signal line **288**.

When manual range control **254** is in neutral range position, solenoid valves C and D are respectively energized and denergized as in forward range position, so forward signal valve **128** remains in the forward position and functions as described above for forward position. In this neutral range position of manual range control **254**, the energization of solenoid valves E and F is reversed—E being deenergized and F energized—so reverse pressure chamber **296'** is pressurized and forward pressure chamber **294'** is exhausted, so reverse signal valve **129** is moved to reverse position. Thus forward signal valve **128** in forward position provides the above-described connections. Reverse signal valve **129** in reverse position blocks forward feed line **299** at land c of spool **291'**, connects forward signal line **288** to exhaust **307** and connects reverse signal line **289** to reverse feed line **301** for connection by forward signal valve **128** to reverse exhaust **302**, so both the forward and reverse signal lines **288** and **289** are exhausted in neutral manual range.

In pivot steer manual range position of manual range control **254**, the same solenoid valves are energized to provide the same operation exhausting both forward and reverse signal lines **288** and **289** as in neutral manual range position described above.

When manual range control **254** is in reverse position, solenoid valves C and E are deenergized to pressurize reverse chambers **296** and **296'**, and solenoid valves D and F are energized to exhaust forward chambers **294** and **294'** so both forward and reverse signal valves **128** and **129** are in reverse position. In forward signal valve **128**, mainline **112** is connected to reverse feed line **301** and both forward feed line **299** and forward chamber feed line **298** are connected to forward exhaust **297**. In reverse signal valve **129**, forward signal line **288** and forward chamber feed line **308** are connected to exhaust **307** and reverse feed line **301** is connected to supply mainline pressure to reverse signal line **289**.

In 2nd and 3rd ratio ranges, inhibitor valve **126** exhausts 3rd chamber feed line **285** so the reverse pressure chambers **296** and **296'** cannot be pressurized—even though solenoid valves C and E are deenergized—so both forward and reverse signal valves **128** and **129** cannot be shifted to reverse position. Thus during operation in forward manual range, 2nd or 3rd ratio range, a shift to neutral, pivot steer, or reverse manual ranges will be inhibited until ratio range is reduced to 1st ratio range, where shifts between all manual ranges are permitted.

In neutral and pivot steer manual range positions, forward signal valve **128** is in forward position and reverse signal valve **129** is in reverse position. On a shift to forward manual range position, only reverse signal valve **129** is shifted to forward position by spring **293'** when reverse pressure chamber **296'** is exhausted by solenoid valve E, and then forward signal pressure from forward signal line **288** via forward chamber feed line **308** pressurizes forward chamber **294'** as solenoid valve F is deenergized and closed. On a shift to reverse manual range, only forward signal valve **128** is shifted to reverse position by energizing solenoid valve D to exhaust forward chamber **294** and deenergizing solenoid valve C to pressurize reverse pressure chamber **296**. On a direct shift from forward to reverse manual range positions, quickly through neutral, both solenoid valves D and F are energized to exhaust forward pressure chambers **294** and **294'** and both solenoid valves C and E are deenergized to pressurize reverse pressure chambers **296** and **296'** to shift both forward and reverse signal valves **128** and **129** to reverse position to supply reverse signal line **289**. On a reverse to forward manual range shift, both soleniod valves C and E are energized to exhaust reverse pressure chambers **296** and **296'** so springs **293** and **293'** shift both forward and reverse signal valves **128** and **129** to forward position to supply forward signal pressure to forward signal line **288** and pressurize forward pressure chambers **294** and **294'** as solenoid valves D and F are deenergized.

DISCONNECT VALVE

Disconnect clutch valve **125** (FIG. 3B) has a spool **314** having three equal-diameter lands a, b, and c, slidable in a bore **316**. At one end of disconnect valve **125**, spring **317** in disconnect chamber **318** and pressure in disconnect chamber **318** may both act on the end of land a of spool **314** to bias spool **314** to disconnect position shown. At the other end of disconnect valve **125**, pressure in connect chamber **319** may act on the end of land c of spool **314** in the opposite direction to the connect position. Disconnect valve **125**, with spool **314** in the disconnect position shown, connects disconnect clutch **24** via disconnect clutch apply line **321** to release disconnect clutch **24**, connects mainline **112** to disconnect chamber feed line **323** which has restriction **324**, and connects mainline **112** to steer signal line **326**. In connect position of spool **314**, disconnect valve **125** connects mainline **112** to disconnect clutch apply line **321** to apply disconnect clutch **24** and connects disconnect chamber feed line **323** and steer signal line **326** to exhaust **327**. Mainline **112** is connected by connect chamber feed line **328** which has restriction **329**, and downstream thereof is controlled by solenoid valve G to control pressure in connect chamber **319**. Disconnect chamber feed line **323**, downstream of its restriction **324**, is controlled by solenoid valve H and supplies disconnect chamber **318**.

When the manual range control **254** is in neutral position, solenoid valve G is energized to exhaust connect chamber **319** so spring **317** moves spool **314** to the disconnect position shown (FIG. 3B), and then mainline

112 feed disconnect chamber feed line 323 to pressurize disconnect chamber 318, as solenoid valve H is deenergized, and feed steer signal line 326. In the other manual range positions (forward, pivot steer, and reverse), solenoid valve G is deenergized to pressurize connect chamber 319 and solenoid valve H is energized to exhaust disconnect chamber 318; so disconnect valve 125, with its spool 314 in connect position, connects mainline 112 to clutch apply line 321 to apply disconnect clutch 24, and connects steer signal line 326 and disconnect chamber 318 and its feed line 323 to exhaust 327. As pointed out below, steer signal line 326 prevents steering in neutral.

ADVANCE RETURN SIGNAL SYSTEM

Neutral Shuttle Valve

The return line 205 is supplied by advance return signal system 160 (FIG. 3C).

The neutral advance line 264 is supplied by neutral shift valve 127 in neutral position, in neutral and pivot steer manual ranges, from advance line 204 supplied by advance return signal system 160. Return line 205 and neutral advance line 264, which alternatively have return signal pressure or neutral advance signal pressure, are connected to neutral shuttle valve 256 (FIG. 3B), which connects the one of return line 205 or neutral advance line 264 having a signal pressure to 1-2-3 return line 331.

RETURN SIGNAL CONTROL VALVE

The return signal control valve 332 (FIG. 3B) has a spool 333 with three equal-diameter lands a, b, and c, slidable in a bore 334 and biased to 1st or low position shown by spring 335 engaging land c of spool 333 and located in chamber 336 in one end of bore 334 vented by exhaust 337. The 2–3 feed pressure supplied by 2–3 feed line 283 in 2nd and 3rd ratio ranges to operating chamber 338 in the opposite end of bore 334 at the opposite end of spool 333 to act on land a of spool 333 to move spool 333 against spring 335 to 2–3 position. With return signal control valve 332 and its spool 333 in 1st or low position shown, 1-2-3 return line 331 is connected to 1st or low return line 339 and 2-3 return line 341 is connected to exhaust 342. With return signal control valve 332 and its spool 333 in 2-3 position, 1-2-3 return line 331 is connected to 2-3 return line 341 and 1st return line 339 is connected to exhaust 343. The 1st or low return line 339 is supplied in neutral or pivot steer manual ranges by neutral advance line 264 or return line 205 and thus has pressure at all times and is supplied in forward and reverse manual ranges only in 1st and reverse ratio drive only with return signal pressure from return line 205.

ADVANCE SIGNAL CONTROL VALVE

The advance signal control valve 344 (FIG. 3B) has a spool 346 having three equal-diameter lands a, b, and c, and a piston 347 of the same diameter slidable in a bore 348. Piston 347 is engageable with land a of spool 346. A spring 349, seated on one end wall 350 of bore 348 and located in spring chamber 351 in bore 348 and vented by exhaust 352, engages land c of spool 346 to bias spool 346 to engage piston 347 which abuts the other or opposite end wall 353 of bore 348 when advance signal control valve 344 and its spool 346 are in the 1–3 position shown (FIG. 3B) in 1st and 3rd forward ratio ranges.

Reverse signal line 329 supplies reverse signal pressure to reverse chamber 354 in bore 348 between piston 347 and other end wall 353. The 2nd clutch line 356, which supplies 2nd clutch pressure to engage 2nd clutch 88, is connected to 2nd clutch chamber 357 in bore 348 between piston 347 and land a of spool 346. When the propulsion drive 13 is operating in reverse drive or 2nd forward ratio range drive, and thus either reverse signal line 289 supplies pressure to reverse chamber 354 or 2nd clutch line 356 supplies pressure to 2nd clutch chamber 357, the spool 346 is moved against spring 349 to R-2nd position.

The advance signal control valve in 1–3 position shown (FIG. 3B), connects 2–3 return line 341 to 2nd-minus ratio change feed line 358 and connects automatic advance line 266 to one branchline 359 of 1st ratio change feed line 361. The other branchline 360 of 1st-plus ratio change feed line 361 is blocked. In the R-2nd position, the advance signal control valve 344 reverses the connections and connects 2-3 return line 341 to the other branchline 360 and 1st ratio change feed line 361 and connects automatic advance line 266 to 2nd ratio change feed line 358. The one branchline 359 of 1st ratio change feed line 361 is blocked.

The 1st and 2nd ratio change feed lines 361 and 358 respectively provide a plus signal for changing displacement from maximum negative displacement toward or to maximum positive displacement in H1 or pump unit 41 to a lower negative displacement in H2 or motor unit 42, and a minus signal for oppositely changing displacement from any value toward or to maximum negative displacement in H1 and H2 units 41 and 42.

As shown in the following TABLE II, the 2nd ratio change feed line 358 provides the 2-3 return signal (R) in 3rd ratio range and the automatic advance signal (A) in Reverse and 2nd ratio ranges. The 1st ratio change feed line 361 provides the 2-3 return signal (R) in reverse and 2nd ratio ranges and the automatic advance signal (A) in 1st and 3rd ratio ranges.

TABLE II

Manual Ranges	Ratio Ranges	Ratio Change Feed Lines 1st	2nd
Fwd	1	A	O
Fwd	2	R	A
Fwd	3	A	R
Rev	Rev	R	A
N	N	O	O

ZERO STROKE BIAS VALVE

The zero stroke bias valve 363 (FIG. 3A) is similar to the above U.S. Pat. No. 3,590,577 and has a valve spool 364 with two equal-diameter lands a and b slidable in a bore 365 in a valve sleeve 366. The spool 364 is connected by linkage 367, which includes a pin 368 and adjustment device 369, to pump ring 370. The pin 368 is secured to ring 370, which is the bearing ring and displacement change member moving proportionally to displacement, of known pump or H1 hydrostatic unit 42 of propulsion hydrostatic transmission 39.

The linkage 367 which connects pump ring 370 to valve spool 364 for common movement has adjustment device 369 so the position of the valve spool 364 can be manually adjusted to the proper position relative to the position of pump ring 370. The adjustment device 369, as shown, has a head apertured to receive pin 368 and a

bolt portion secured to spool **364** and locked in the adjusted position by a locknut.

Valve sleeve **366** has a reverse return port **371** and a forward return port **372** and is slidable in bore **373** in valve body **169**. Spring **374** in spring chamber **376** in valve body **169** is seated on spring seat **375** and engages abutment **380** on one end of valve sleeve **366** and biases valve sleeve **366** to the reverse position shown, with the other end engaging piston **377**, which also is slidable in bore **373** in the valve body **169**, and abuts the end wall **378**.

In forward manual range, forward signal line **288** supplies forward signal pressure to chamber **379** in bore **373** in valve body **169** between end wall **378** and piston **377** and valve sleeve **366** so its opposite end engages spring seat **375** wall with sleeve **366** in forward position. Exhaust **381** vents the bore **365** of valve sleeve **366** between piston **377** and land b of spool **364** and bore **373** in valve body **169**. At the other end of valve sleeve **366** these bores **365** and **373** are vented past spring seat **375** wall.

The 1st or low return line **339**, which in neutral is always supplied since it is fed by both neutral advance line **264** and return line **205** and which in 1st and reverse ratio ranges is supplied only by return line **205**, is connected via supply port **382** to the space between lands a and b of spool **346** for selective connection by reverse return port **371** to reverse return line **383** and 1st forward return port **372** to 1st forward return line **384**. These reverse and 1st forward return ports **371** and **372** are formed in sleeve **366** as a plurality of progressively wider slots with the narrowest slot of each port at the regulating edge respectively of lands a and b of spool **364** so flow to reverse or 1st forward return lines **383** or **384** is proportional to the degree of error or deviation from the desired 1st and reverse ratio range anti-hunting or bias positions so the rate of change of correction is proportional to the error.

In the reverse position of zero stroke bias valve **363** with valve spool **364** in the central blocking position shown, the pump ring **370** is slightly displaced from null (N) or zero stroke position to a reverse bias (RB) position (FIGS. 3A and 4) having very low reverse or negative displacement. In response to a displacement position error due to higher or lower displacement from reverse bias (RB) position, the zero stroke bias valve **363** respectively provides reverse return signal in reverse return line **383** or forward return signal in 1st forward return line **384**, functioning as a reverse advance signal, for correction.

In the forward position, in response to forward signal supplied by forward signal line **288**, the valve sleeve **366** is moved to the left in FIG. 3A to forward position; and when valve spool **364** is in a similar central blocking position relative to reverse and 1st forward return ports **371** and **372**, the pump ring **370** is in forward bias (FB) position (FIGS. 3A and 4), displaced a little from null (N) position, to a very low forward or positive displacement position. On displacement error from forward bias (FB) position to higher or lower displacement positions, zero stroke bias valve **363** respectively provides forward return pressure in 1st forward return line **384** or reverse return signal pressure, acting as forward advance signal pressure in reverse return line **383**. This arrangement provides a damped minimum displacement control to prevent hunting of the displacement control by placing the hydrostatic pump, under low load providing creep torque and conditioning the propulsion drive **13** in forward or reverse manual range for immediate forward 1st ratio range or reverse ratio range drive in response to torque demand.

2-3-2 SHIFT SIGNAL VALVE

The 2-3-2 shift signal valve **386** has a spool **387** having two equal-diameter lands a and b slidable in a bore **388** which is in valve body **169** and is vented at both ends. Linkage **389**, which is like linkage **367** of zero stroke bias valve **363** and has adjustment device **390**, connects spool **387** to pump ring **370** for proportional movement.

The forward signal line **288** is always connected to the space in bore **388** between lands a and b of spool **387**. The 2-3-2 shift signal line **391** is connected to its port **392**, and when pump ring **370** and valve spool **387** are not in 2-3-2 shift position as shown (FIG. 3A), 2-3-2 shift signal line **391** is connected by its port **392** around land a of spool **387** to exhaust line **393** connected to bore **388** beyond land a of spool **387** for exhaust. When pump ring **370** and spool **387** are in the 2-3-2 shift position, land a of spool **387** closes exhaust line **393** and connects forward signal line **288** to supply 2-3-2 shift signal line **391**.

1-2-1 SHIFT SIGNAL VALVE

The 1-2-1 shift signal valve **396** has a valve spool **397** having two equal-diameter lands a and b slidable in bore **398** and connected by linkage **399**, like the above linkage **367**, to motor ring **401**. Bore **398** is vented at both ends. When motor ring **401** and thus valve spool **397** (as shown in FIG. 3A) of 1-2-1 shift signal valve **396** are in other than 1-2-1 shift position, 1-2-1 shift signal valve **396** connects 1-2-1 shift signal line **402** to exhaust **403**. When motor ring **401** and 1-2-1 shift signal valve **396** are in 1-2-1 shift position, 1-2-1 shift signal valve **396** connects forward signal line **288** to supply 1-2-1 shift signal line **402**.

ZERO STROKE INDICATOR VALVE

The zero stroke indicator valve **132** (FIG. 3D) has a spool **407** having two equal-diameter lands a and b slidable in a bore **408** vented at both ends thereof by 1st and 2nd exhausts **413** and **413'**. Linkage **409**, like linkage **367**, connects spool **407** to pump ring **370**. Mainline **112** is always connected to bore **408** between lands a and b of spool **407** and is blocked so there is only leakage flow to 1st and 2nd ports **410** and **410'** in the zero or null (N) stroke position of spool **407** as shown in FIG. 3D. This leakage flow past lands a and b of spool **407**, respectively, to 1st and 2nd ports **410** and **410'** is permitted to escape at low pressure via 1st and 2nd restricted passages **412** and **412'** and extending respectively from 1st and 2nd ports **410** and **410'** beyond lands a and b of spool **407** and returning to bore **408** for exhaust by exhausts **413** and **413'** at opposite ends of bore **408**. This arrangement reduces leakage and the low pressure produced in zero stroke feed lines **411** and **411'** is not significant.

When pump ring **370** moves from zero stroke or null (N) position in either direction, spool **407** moves to connect mainline **112** to selectively supply either of 1st or 2nd ports **410** or **410'** and 1st and 2nd zero stroke feed lines **411** and **411'**. After movement of ring **370** in one direction from zero stroke position (N), with land a of spool **407** moving to connect mainline **112** to 1st port **410** and 1st zero stroke feed line **411** and 1st restricted passage **412**, with small further movement land a of spool **407** blocks the connection of 1st restricted passage

to 1st exhaust **413** to close the leakage path, and land b of spool **407** has a large overlap of 2nd port **410'** to prevent leakage. When pump ring **370** moves from zero stroke (N) position in the other or opposite direction, mainline **112** is similarly connected to oppositely disposed and similar 2nd port **410'**, 2nd zero stroke feed line **411'**, and then 2nd restricted passage **412'** and 2nd exhaust **413'** is closed by land b of spool **407**, and the large overlap of land a of spool **407** relative to 1st port **410** reduces leakage. The 1st and 2nd zero stroke feed lines **411** and **411'** are selectively supplied and when either one is supplied with zero stroke indicator signal pressure, the zero stroke shuttle valve **414** connects the zero stroke indicator signal pressure to zero stroke indicator signal line **416** whenever pump ring **370** is not in zero stroke (N) position. Thus zero stroke signal pressure at zero pressure provides the zero stroke signal.

ADVANCE TIME DELAY DEVICE

The advance time delay device **417** (FIG. 3B) has a piston **418** slidable in a bore **419** vented by exhaust **426** between piston **418** and end wall **422**. Spring **421** seated on end wall **422** engages and biases piston **418** to the discharged position shown. In discharged or exhaust position, automatic advance line **266** is exhausted at neutral shift valve **127** and delayed automatic advance line **423** is connected by both its supply port **424** and exhaust port **425** to bore **419** and via bore **419** to exhaust **426**. When neutral shift valve **127** supplies automatic advance line **266**, automatic advance signal pressure is supplied via automatic advance line **266** through restriction **427** to expanding chamber **428** in bore **419** between piston **418** and end wall **430** to move piston **418** against the bias force of spring **421** during a predetermined time period to supply position. In this supply position, advance time delay device **417** connects automatic advance line **266** through restriction **427** and expanding chamber **428** to supply port **424** and delayed automatic advance line **423** and blocks exhaust port **425**. When the supply port **424** is fully open, piston **418** engages stop **429** and exhaust **426** continues to vent the bore **419** between piston **418** and end wall **422**.

2-3 OVERPRESSURE CUT-OFF VALVE

The 2-3 overpressure cut-off valve **431** (FIG. 3F) has a spool **432** having five equal-diameter lands a, b, c, d, and e, and a piston **433** of the same diameter, both slidable in a bore **434**. A spring **436** seated on one end wall **437** in a spring chamber **438** vented by exhaust **439**, engages land e of spool **432** to bias spool **432** so land a of spool **432** abuts piston **433** which, in turn, abuts the other end wall **441** in the normal position shown. In this normal position, the 2-3 overpressure cut-off valve **431** connects range feed line **267** between lands c and d of spool **432** to 1st range supply line **442** connects overpressure signal line **252** between lands d and e of spool **432** to exhaust **443** and blocks lock signal line **444** between lands a and b of spool **432**. Also, lock chamber **446** between piston **433** and land a of spool **432** is connected to exhaust **448**. Thus in normal position, the 2-3 overpressure cut-off valve **431** functions only to connect range feed line **266** to 1st range supply line **442**.

The output or 2nd governor signal pressure line **157** and zero stroke indicator signal line **416** are connected to lock shuttle valve **445** which supplies the higher of 2nd governor pressure and zero stroke indicator signal pressure to lock signal line **444**. When there is high hydrostatic overpressure as described below, high pressure safety signal valve **451** (FIG. 3A) connects mainline pressure from 2-3 feed line **283** to high pressure signal line **452**. High pressure signal line **452**, in 2-3 overpressure cut-off valve **431**, is connected to high pressure signal chamber **453** between end wall **441** and piston **433** to act on piston **433** and move piston **433** and spool **432** against spring **436** to the cut-off position with land e of spool **432** abutting stop **454**. The 2-3 overpressure cut-off valve **431** is thus moved to cut-off position only in 2nd and 3rd ratio ranges when there is a high hydrostatic overpressure.

The 2-3 overpressure valve **431**, in cut-off position, provides the following connections: Range feed line **267** is connected between lands c and d of spool **432** to overpressure signal line **252** which is connected by return demand shuttle valve **251** to supply return demand signal line **227** so advance return signal system **160** only provides a return signal to return line **205**, as explained above. The 1st range supply line **442** is connected between lands b and c of spool **432** to exhaust **448** to disestablish all ratio ranges, 1st brake **65**, 2nd clutch **88**, and 3rd clutch **72**. The lock signal line **444** is connected between lands a and b of spool **432** to lock transfer line **447** to supply lock chamber **446** to lock 2-3 overpressure cut-off valve **431** in cut-off position until the high pressure signal in high pressure signal line **452** is discontinued indicating proper high hydrostatic pressure, until the zero stroke indicator signal pressure line **416** is exhausted indicating hydrostatic pump **H1** is at substantially zero displacement, a displacement not greater than forward and reverse bias positions (FB), (RB), and until output governor pressure in 2nd governor pressure line **147** and vehicle speed are very low (e.g., 5 mph), so 1st ratio range drive can be smoothly reestablished.

RATIO RANGE SHIFT CONTROLS

Priority Valve

The ratio range shift controls **460** (FIG. 3E) are supplied by 1st range supply line **442** which is connected to priority valve **456** to immediately with priority, supply 2nd range supply line **457** and after mainline supply requirements are met, to supply 3rd range supply line **458**.

Priority valve **456** has a valve spool **459**, slidable in a bore **461** and biased by a spring **462** in a spring chamber **463** vented by exhaust **464**, to the closed position shown connecting 1st range supply line **442** directly for full-flow to 2nd range supply line **457** and through restriction **466** to 3rd range supply line **458**. Since 2nd range supply line **457** is only used as a hold-pressure in 1-2 shift valve **491** as explained below, its flow requirements are not significant. When the flow requirements of mainline **112** are met at a proper operating pressure (somewhat less than mainline pressure), priority valve **456** opens to provide a full-flow connection to 1st range supply line **442**, used for all ratio range establishing supply and exhaust pressure control during ratio range disestablishing.

RATIO SHIFT SIGNAL VALVE

The ratio shift signal valve **257** (FIG. 3E) has a valve spool **468** having five equal-diameter spaced lands a, b, c, d, and e, slidably disposed in a valve bore **469**, 1st and 2nd springs **471** and **472** seated on opposite ends of valve bores **469** and centering the valve spool **468** in valve bore **469**. Ratio shift signal valve **257** also has 1st

and 2nd pistons 473 and 474 of smaller diameter than spool 468 and slidably disposed in series in 1st piston bore 476 at one end of valve bore 469, and 3rd and 4th pistons 477 and 478 of the same smaller diameter slidably disposed in 2nd piston bore 479 at the other end of valve bore 469. The 1st and 2nd piston bores 476 and 479 have their outer ends closed and their central ends open to valve bore 469 so 2nd piston 474 can move within 1st spring 471 and engage land a of spool 468, and 3rd piston 477 can move within 2nd spring 472 and engage land e of spool 468.

The return chamber 481 in the 1st end of valve bore 469 containing 1st spring 471 is connected to return line 205, and advance chamber 482 in the 2nd end of valve bore 469 containing 2nd spring 472 is connected to delayed automatic advance line 423. When a return signal is present, valve spool 468 will be shifted to the right against 2nd spring 472 to downshift position, allowing a downshift signal from the ratio shift signal valve 257. When an advance signal is present, valve spool 468 will be shifted to the left against 1st spring 471 to upshift position allowing an upshift signal from ratio shift signal valve 257.

Hysteresis or damping is obtained by the rate of ratio change controls 203 and advance time delay device 417 which delays the engine overspeed or advance signal to upshift and promptly provide the engine underspeed or return signal to downshift. The valve bore 469 is connected with the following lines and exhausts which are located relative to valve spool 468 in the central position as follows. The 2-3-2 and 1-2-1 shift signal lines 391 and 402 are respectively blocked by lands b and d of spool 468. The exhausts 483, 484, and 485 are respectively blocked by lands a, c, and e of spool 468. The 3-2, 2-3, 2-1, and 1-2 shift signal lines 486, 487, 488, and 489, are respectively connected between successive lands, from land a to land e of spool 468.

The 2-1 shift signal line 488 is connected to the closed end of 1st piston bore 476 adjacent the outer end of 1st piston 473 and to 1-2 shift valve 491. The 3-2 shift signal line 486 is connected to 1st piston bore 476 between the 1st and 2nd pistons 473 and 474. The 1-2 shift signal line 489 is connected to 2nd piston bore 479 between 3rd and 4th pistons 477 and 478. The 2-3 shift signal line 487 is connected to the closed end of 2nd piston bore 479 adjacent the end of 4th piston 478. The 1-2 shift signal line 489 is connected to 1-2 shift valve 491, and the 3-2, 2-3, and 2-1 shift signal lines 486, 487, and 488 are connected to the 2-3 shift valve 492 to control the shift valves as explained below.

1-2 SHIFT VALVE

The 1-2 shift valve 491 (FIG. 3E) has a valve spool 493 having four spaced lands a, b, c, and d, and piston 494 of equal diameter, both slidably disposed in valve bore 496. A spring 497 is compressed in downshift chamber 498 in valve bore 496 between one end thereof and land a of spool 493. Spring 497 and pressure in downshift chamber 498 supplied by 2-1 shift signal transfer line 514 act on land a of spool 493 to bias spool 493 to 1st position shown. Pressure supplied by 1-2 shift signal line 489 to 1st upshift chamber 501 at the other end of bore 496 between piston 494 and spool 493, and pressure supplied by transfer line 506 to 2nd upshift chamber 502 between piston 494 and the end of bore 496, both bias spool 493 and 2nd position. The valve bore 496 is successively connected to and controls 1st exhaust line 503, 1st apply line 239, 3rd range supply line 458, 2-3 feed line 283, 2nd exhaust line 504, 2nd range supply line 457, transfer line 506, and exhaust 507. In 1st or downshift position shown, 3rd range supply line 458 is connected to 1st apply line 239 so 1st brake 65 or the 1st forward and reverse ratio friction establishing device, will be engaged. Also, the 2nd upshift chamber 502 is connected, via transfer line 506 and between lands c and d of spool 493, to exhaust 507, 2nd range supply line 457 is blocked by land c of spool 493, and 2-3 feed line 283 is connected between lands b and c of spool 493 to 2nd exhaust line 504. When a shift signal is present in the 1-2 shift signal line 489 and 1st upshift chamber 501, the valve spool 493 is moved to the left to 2nd or upshift position to connect 3rd supply line 458 to 2-3 feed line 283 and 1st apply line 239 to 1st exhaust line 503. Also the 2nd range supply line 457 is connected via transfer line 506 to 2nd upshift chamber 502 to act on piston 494 to hold valve spool 493 in the upshift position so that the 1-2 shift signal in line 489 does not have to be maintained after the 1-2 shift has been completed. The 1st exhaust line 503 is connected with 1st and 2nd trip valves 508 and 509, and the 2nd exhaust line 504 is connected with 2nd and 3rd trip valves 509 and 511, as described below.

2-3 SHIFT VALVE

The 2-3 shift valve 492 has the same structure as the 1-2 shift valve 491, so like reference numerals (primed) are used and the difference connections provided are now described.

The valve bore 496' (FIG. 3E) is successively connected to and controls 2nd exhaust line 504, 2nd apply line 356, 2-3 feed line 283, 3rd apply line 512, 3rd exhaust line 513, exhaust 516, 2-1 shift signal transfer line 514, and 2-1 shift signal line 488. In 2nd or downshift position shown, the 2-3 feed line 283 is connected to 2nd apply line 356 and 2nd exhaust line 504 is blocked so that if 1-2 shift valve 491 is upshifted, 2nd clutch 88 (the 2nd ratio engaging device) will be engaged. Also, 3rd apply line 512 is connected to 3rd exhaust line 513, and 2-1 shift signal line 488 is connected to 2-1 shift signal transfer line 514 which supplies downshift chamber 498' for downshift of 1-2 shift valve 491. When 2-3 shift signal line 487 supplies, from ratio shift signal valve 257, a 2-3 shift signal to 1st upshift chamber 501', valve spool 493' will be moved to the left against spring 497' to 3rd or upshift position, thereby connecting 2-3 feed line 283 to 3rd apply line 512 to engage 3rd clutch 72 and to supply 2nd upshift chamber 502' to maintain 2-3 shift valve 492 in the upshift position connecting 2nd apply line 356 to 2nd exhaust line 504 to disengage 2nd clutch 88. Also, the 2-1 shift signal line 488 is blocked and 2-1 shift signal transfer line 514 is connected to exhaust 516, so 1-2 shift valve 491 cannot be downshifted when 2-3 shift valve 492 is upshifted. The 3-2 shift signal line 486 is connected to downshift chamber 498' for downshift of 2-3 shift valve 492. The 3rd exhaust line 513 is connected to 2nd and 3rd trip valves 509 and 511.

TRIP VALVES

The 1st trip valve 508 (FIG. 3E) has a valve spool 518 having two lands a and b and a piston 519 of the same diameter slidably disposed in a valve bore 521. A spring 522 is compressed in vented chamber 523 and located between one end of valve bore 521 and land a of spool 518 to urge the valve spool 518 and piston 519 to the feed position shown in which 3rd range supply line 458 is connected by restricted feed line 525 to supply valve

bore **521** between lands a and b of spool **518**. Fluid pressure in a 1st chamber **526** between land b of spool **518** and piston **519** and also in a 2nd chamber **527** between piston **519** and the end wall of valve bore **521** act to move valve spool **518** against spring **522** to the exhaust position.

The 2nd and 3rd trip valves **509** and **511** have the same structure as 1st trip valve **508**, so like reference numerals for the parts, respectively single and double-primed have been used with reference to be above description of 1st trip valve **508**, and the connections are now described.

The 3rd range supply line **458** is connected to each restricted feed line **525**, **525'**, and **525''**, respectively, of 1st, 2nd, and 3rd trip valves **508**, **509**, and **511** and also is connected via restriction **528** to low pressure exhaust line **529** which is regulated at a low pressure by relief valve **531**. Low pressure exhaust line **529** is connected to each of 1st, 2nd, and 3rd trip valves **508**, **509**, and **511** and blocked respectively from bores **521**, **521'**, and **521''** by lands a, a' and a'' of spools **518**, **518'**, and **518''**. The 1st, 2nd, and 3rd exhaust lines **503**, **504**, and **513**, are respectively connected to bores **521**, **521'**, and **521''**, between lands a and b, a' and b', and a'' and b'', of 1st, 2nd, and 3rd trip valves **508**, **509**, and **511**, and respectively by restrictions **532**, **533**, and **534**, to low pressure exhaust line **529**. In 1st trip valve **508**, 2nd and 3rd apply lines **356** and **512** are respectively connected to 1st and 2nd chambers **526** and **527**. In 2nd trip valve **509**, 1st and 3rd apply lines **239** and **512** are respectively connected to 1st and 2nd chambers **526'** and **527'**. In 3rd trip valve **511**, 1st and 2nd apply lines **239** and **356** are respectively connected to 1st and 2nd chambers **526''** and **527''**.

The 1st trip valve **508** in the feed position shown during 1st ratio range drive, so on a 1-2 shift 1st exhaust line **503** is initially connected via restricted feed line **525** to 3rd range supply line **458** and via restriction **532** to low pressure exhaust line **529** so that initially on an upshift of the 1-2 shift valve **491**, the 1st brake **65** will remain pressurized until 2nd clutch **88** reaches a predetermined pressure and torque transmitting level. Restriction **532** to low pressure exhaust line **529** limits the pressure in 1st exhaust line **503**. When the pressure level in 2nd clutch **88** increases to this predetermined pressure, it is connected via 2nd apply passage **356** to 1st chamber **526** to move valve spool **518** against spring **522** to the left to the exhaust position. The 1st trip valve **508**, in exhaust position, blocks restricted feed line **525** and connects 1st exhaust line **503** via valve bore **521** between lands a and b of spool **518**, bypassing restriction **532** directly to low pressure exhaust line **529** to exhaust 1st brake **65**. The 1st trip valve **508** will be similarly operated on a 1-3 shift when 3rd clutch **72** is engaged.

With the 2nd trip valve **509** in feed position shown, during 2nd ratio range drive, the 3rd range supply line **458** is connected through restricted feed line **525'** to 2nd exhaust line **504** which is also connected by restriction **533** to low pressure exhaust line **529**. On a 2-3 upshift to 3rd ratio range or a 2-1 downshift to 1st ratio range, 2nd exhaust line **504** will remain pressurized until the pressure in 3rd or 1st apply line **512** or **239** is sufficient to engage the 3rd clutch **72** or 1st brake **65** and to overcome spring **522'** to shift valve spool **518'** to block restricted feed line **525'** and to connect 2nd exhaust line **504** for free flow to low pressure exhaust line **529** to disengage 2nd clutch **88**. The 3rd trip valve **511**, as shown, is in feed position during 3rd ratio range drive, so that 3rd range supply line **458** is connected via restricted feed line **525''** to 3rd exhaust line **513** to maintain 3rd clutch **72** engaged during the initial portion of a downshift from 3rd ratio range until 2nd or 1st ratio range is engaged, as described above for the 2nd trip valve **509**.

PROPULSION HYDROSTATIC TRANSMISSION

Propulsion hydrostatic transmission **39** has known variable displacement pump or H1 and motor or H2 hydrostatic units **41** and **42** respectively having pump and motor rings **370** and **401** pivotally supported by pins **535** and **536** mounted on housing **38** (FIG. 3A).

The pump and motor rings **370** and **401** respectively have a zero displacement or null positions (PN) and (MN). During drive operation, the pump ring **370** passes through null position (PN), but the motor ring **401** is not used in null position (MN).

The hydrostatic H1 unit or pump **41** has 1st or plus and 2nd or minus pump servos **537** and **538**, piston and cylinder type fluid motors, which are fixed on housing **38** and operatively connected by their piston rods to pump ring **370** to change displacement. The 1st or plus pump servo **537** moves pump ring **370** in a positive direction toward or to maximum positive displacement position (P+) for 1st and 3rd ratio range drives **1,3**. The 2nd or minus pump servo **538** moves pump ring **370** in a negative direction toward or to 2nd ratio range position **2**, and then to maximum negative displacement position (P−) for reverse ratio range drives, as shown in FIG. 3A.

The hydrostatic motor **42** has similar 3rd or plus and 4th or minus motor servos **539** and **540** which are connected to motor ring **401** to move it respectively in a positive direction to an intermediate negative displacement position (MI), and in a negative direction to a maximum negative displacement position (M−), as shown in FIG. 3A.

The propulsion hydrostatic transmission **39** has known 1st and 2nd kidney passages **541** and **542** connecting pump and motor hydrostatic units **41** and **42**, one carrying high hydrostatic pressure and the other carrying low hydrostatic pressure, depending on the direction of power flow, for transmitting power between the pump and motor units **41** and **42**.

RATIO CHANGE ACTUATOR

The ratio change actuator **543** (FIG. 3B) is fixed on housing **38** and has an actuator piston **544** slidable in cylinder **545** which has 1st and 2nd end walls **546** and **547**. Piston **544** has a rod **548** slidably sealed in an aperture in 1st end wall **546**. A stop piston **551**, slidable in a control cylinder **552**, has a stop member **553** slidably sealed in an aperture in 2nd end wall **547** and extending into cylinder **545** to provide a two-position stop for actuator piston **544**. Control cylinder **552** extends from 2nd end wall **547** to a 3rd end wall **555**. An exhaust **554** vents the space in control cylinder **552** between stop piston **551** and 2nd end wall **547**. In forward manual range, forward signal line **288** supplies forward signal pressure to the chamber **556** in control cylinder **552** between 3rd end wall **555** and stop piston **551** to act on stop piston **551** to position stop member **553** in cylinder **545** to limit movement of actuator piston **544** to 2nd and 3rd ratio maximum negative position **2**. The actuator piston **544** is shown in the central null (AN) position.

When 2nd or minus actuator pressure is supplied by 2nd actuator line **558**, actuator piston **544** is moved in forward range toward and to 2nd position **2**, against

stop member 553 in forward position shown, and in reverse range drive, further to reverse position (R), FIG. 3B. When 1st or plus actuator pressure is supplied by 1st actuator line 557, actuator piston 544 is moved toward and to the 1st and 3rd ratio range forward positions 1, 3, (FIG. 3B).

As pointed out above, advance signal control valve 344 supplies 1st (plus) and 2nd (minus) ratio change feed lines 361 and 358. Also, zero stroke bias valve 363 supplies reverse (minus) and forward (plus) return lines 383 and 384. The 1st or plus shuttle valve 561 connects the pressurized one of 1st or plus ratio change feed line 361 or forward return (plus) line 384 to 1st or plus actuator line 557. The 2nd or minus shuttle valve 562 connects the pressurized one of 2nd or minus ratio change feed line 358 or reverse return (minus) line 383 to 2nd or minus actuator line 558. The 1st and 2nd actuator lines 557 and 558 have advance (AP) and return (RP) pressures, as shown in TABLE III.

TABLE III

DRIVE RANGES		ACTUATOR LINES 1st Plus (557)	2nd Minus (558)
Manual	Ratio		
Fwd.	1st	AP	Rev. RP
Fwd.	2nd	RP	AP
Fwd.	3rd	AP	RP
Rev.	1st	Fwd. RP	AP
Neutral	N	Fwd. RP	Rev. RP

In TABLE III, forward 1st and reverse, the signal pressures supplied are shown for operation above idle throttle when ratio governor 162 may provide advance pressure. At idle throttle in forward 1st and reverse, the signal pressures supplied are the same as in neutral under the control of zero stroke bias valve 363 to provide forward bias position (FB) in forward 1st and reverse bias position (RB) in reverse and neutral as explained above.

SERVO VALVES

The pump servo valve 563 (FIG. 3A) is a known rotary type having a spool 566, with four axial grooves arranged as two pairs of opposite grooves, and is rotatably mounted in a sleeve 568 having ports and being rotatably mounted in valve body 169. Pump servo valve 563 is used to deliver control pressure fluid to vary the displacement of H1 or pump hydrostatic unit 41. Control pressure line 564 delivers control pressure, regulated by propulsion control pressure regulator valve 138, through the ports in sleeve 568 to one pair of grooves in spool 566. The other pair of grooves are connected to exhaust 565.

The spool 566 is connected by linkage 567 for rotation by movement of actuator piston 544, and sleeve 568 is connected by linkage 569 for rotation by movement of pump ring 370. The pump servo valve 563 selectively delivers control pressure to 1st (plus) and 2nd (minus) pump control feed lines 571 and 572 which are respectively connected through overpressure system 573 (described below), to 1st and 2nd pump control supply lines 574 and 575 to 1st and 2nd pump servos 537 and 538.

The pump servo valve 563 functions such that as spool 566 is rotated clockwise, 2-Rev. movement, control pressure is distributed from control pressure line 564 through 2nd or minus control feed line 572 to control 2nd pump servo 538 which will cause the pump 41 to change displacement toward 2-R, or negative displacement position (P−). As the pump 41 changes displacement, the sleeve 568 rotates with movement of pump ring 370 to cause sleeve 568 to rotate clockwise to block this control pressure flow to 2nd pump servo 538 when the desired displacement has been reached. If spool 566 is rotated counterclockwise, 1, 3 movement, control pressure will be distributed to 1st pump (minus) servo control feed line 571 to control 1st pump servo 537 to cause the pump 41 to change displacement in the opposite direction toward 1, 3 position, the maximum positive displacement position (P+). Again, sleeve 568 will be driven by pump ring 370 until the desired displacement is achieved, and the pressure connections between control pressure line 564 and 1st pump control feed line 571 are closed.

The motor servo valve 576 (FIG. 3A) is like pump servo valve 563, so the same reference numerals (primed) have been used and reference is made to the above description of pump servo valve 563.

In motor servo valve 576, spool 566′ is connected by linkage 578 to linkage 567 which is moved by actuator piston 544 to rotate spool 566 of pump servo valve 563 between an intermediate negative displacement position (MI−) and a maximum negative displacement position (M−).

In motor servo valve 576, the sleeve 568′ is connected by linkage 578 to rotate with displacement change movement of motor ring 401. In response to movement of actuator piston 544, the spool 566′ is moved in a negative or positive direction to selectively connect control pressure line 564 to 1st and 2nd motor control supply lines 579 and 580, respectively, connected to 1st (plus) and 2nd (minus) motor servos 539 and 540 of motor hydrostatic unit 42 to move motor ring 401 toward maximum negative displacement position (M−), or toward intermediate negative displacement position (MI−).

PROPULSION HYDROSTATIC CHARGING SYSTEM

In the propulsion hydrostatic charging system 134 (FIG. 3A), mainline 112 is connected via 1st and 2nd check valves 581 and 582 to feed charging fluid to the one of 1st or 2nd kidney feed lines 583 or 584 and connected 1st or 2nd kidney passages 541 or 542 having low hydrostatic pressure. The 1st and 2nd kidney feed lines 583 and 584 also supply 1st and 2nd kidney pressures to 1st and 2nd kidney pressure signal lines 586 and 587.

OVERPRESSURE SYSTEM

In the overpressure system 573 (FIG. 3A) there is a propulsion stroke limiter valve 588 which connects 1st and 2nd pump control feed lines 571 and 572 to 1st and 2nd pump control connecting lines 589 and 590, which are connected by safety mode valve 591 respectively to 1st and 2nd pump control supply lines 574 and 575 for supplying regulated control pressure to 1st and 2nd pump servos 537 and 538 to control displacement of pump hydrostatic unit 41 only when the high kidney pressure is in a normal pressure range. When the high kidney pressure is higher (overpressure), the propulsion stroke limiter valve 588 and safety mode valve 591 function to decrease high control pressure and thus high hydrostatic kidney pressure.

PROPULSION STROKE LIMITER VALVE

The propulsion stroke limiter valve **588** (FIG. 3A) is a known type stroke limiter valve, like the steer stroke limiter valve **592** described below.

The propulsion stroke limiter valve **588** under overload, higher than normal high kidney pressure, indicative of torque overload in propulsion hydrostatic transmission **39**, acts in response to high kidney pressure supplied by one of 1st or 2nd kidney pressure signal lines **586** or **587** to sequentially limit, interrupt, and then reverse the control pressure supplied to 1st and 2nd pump servos **537** and **539**. Thus the degree of stroke or displacement request by ratio change actuator **543** is overridden to first decrease control pressure permitting a decrease in displacement and then to reverse control pressures for a positive decrease in displacement to reduce high kidney pressure to the normal range.

SAFETY MODE VALVE

The safety mode valve **591** (FIG. 3A) has a spool **593** having five equal-diameter lands a, b, c, d, and e, slidable in bore **594**. Spring **596** is seated in the vented end of bore **594** and engages land a of spool **593** to bias spool **593** to connecting position shown. The 2-3 feed line **283** is connected to chamber **597** at the opposite end of bore **594** to act on land e of spool **593** to move spool **593** against spring **596** in 2nd and 3rd ratio ranges to the safety mode position. In connecting position, safety mode valve **591** connects 1st and 2nd pump control connecting lines **589** and **590** to 1st and 2nd pump control supply lines **574** and **575**. 1st and 2nd control supply branch-lines **598** and **599** are blocked and 1st and 2nd overpressure signal lines **601** and **602** are connected to 1st and 2nd exhausts **603** and **604** so high pressure safety signal valve **451** is closed.

In safety mode position of safety mode valve **591**, 1st and 2nd pump control feed lines **571** and **572** are connected by 1st and 2nd control branchlines **598** and **599** which bypass propulsion stroke limiter valve **588** and are connected through safety mode valve **591** to 1st and 2nd pump control supply lines **574** and **575** for 1st and 2nd pump servos **538** and **539**. 1st and 2nd pump control feed lines **571** and **572** are also connected through propulsion stroke limiter valve **588**, 1st and 2nd pump control connecting lines **589** and **590**, safety mode valve **591** and 1st and 2nd overpressure signal lines **601** and **602** to high pressure safety signal valve **451**.

HIGH PRESSURE SAFETY SIGNAL VALVE

High pressure safety signal valve **451** (FIG. 3A) has a spool **605** having equal-diameter lands a and b slidable in valve bore **606**, and 1st and 2nd pistons **607** and **608** of smaller diameter arranged in series in coaxial smaller diameter piston bore **609** which is closed at its outer end and open to one end of valve bore **606** which is vented by exhaust **613**. Spring **611**, seated on the other vented end of valve bore **606**, engages land b of spool **605** to bias spool **605** to engage 1st piston **607** which, in turn, engages 2nd piston **608** against the closed outer end of piston bore **609** to thereby place high pressure safety signal valve **451** in the closed position shown connecting high pressure signal line **452** to the exhaust **612** and blocking 2-3 feed line **283**.

When a high overpressure signal indicating overpressure is supplied by either of 1st or 2nd overpressure signal lines **601** or **602**, to piston bore **609** between 1st and 2nd pistons **607** and **608**, or to the end of piston bore **609** to act on 1st piston **607** to move valve spool **605** to over-pressure position, 2-3 feed line **283** is connected to high pressure signal line **452** which controls the 2-3 overpressure cut-off valve **431**, as explained above.

CONTROL PRESSURE REGULATION SYSTEM

The control pressure regulation system **616** (FIG. 3A) has hydrostatic control pump **136** supplied by mainline **112** and supplying control pressure line **564** regulated by propulsion control pressure regulator valve **138** which is controlled by high kidney pressure from both the propulsion and steer hydrostatic transmissions **39** and **89**.

PROPULSION SHUTTLE VALVE

The propulsion shuttle valve **617** (FIG. 3A) has a spool **618** having three equal-diameter lands a, b, and c, slidable in a bore **619**. In propulsion pump hydrostatic unit **41**, 1st and 2nd kidney pressure supply lines **621** and **622** supply 1st and 2nd kidney pressures to propulsion shuttle valve bore **619** and also, via 1st and 2nd restrictions **623** and **624**, to 1st and 2nd chambers **626** and **627** at opposite ends of bore **619**. When, as shown, 2nd kidney pressure is higher than 1st kidney pressure, the higher 2nd kidney pressure in 2nd chamber **627** moves valve spool **618** to the position shown and connects 2nd kidney pressure supply line **622** to supply propulsion high kidney pressure line **629**, and the lower kidney pressure in 1st kidney pressure supply line **621** to supply propulsion low kidney pressure line **630**. When 1st kidney pressure is the higher pressure, this pressure in 1st kidney pressure supply line **621** similarly acts in 1st chamber **626** to move valve spool **618** to the opposite end to connect the higher 1st kidney pressure to propulsion high kidney pressure line **629** and the lower 2nd kidney pressure to propulsion low kidney pressure line **630**.

LOW CHARGE SIGNAL VALVE

The propulsion hydrostatic transmission **39** has a low charge signal valve **631** (FIG. 3A) having a spool **632** with equal-diameter lands a and b slidable in a valve bore **633**. Spring **634** in vented chamber **636** is seated on the end of bore **633** and engages land b of spool **632**. Low kidney pressure is supplied by propulsion low kidney pressure line **630** to chamber **637** to act on land a of spool **632** to oppose the spring bias force. When low kidney pressure is abnormally low, spring **634** overcomes the low kidney pressure bias to move spool **632** to the low charging pressure position shown, connecting mainline **112** to underpressure or low charge pressure signal line **253** connected to return demand shuttle valve **251** (FIG. 3F) to supply return demand line **227**, as pointed out above.

When low kidney pressure has normal low pressure in a range of normal low kidney pressure values, this normal pressure in chamber **637** moves valve spool **632** against spring **634** to block mainline **112** and to connect underpressure signal line **253** to exhaust **638**.

1ST OR CONTROL PRESSURE REGULATOR VALVE

The 1st or control pressure regulator valve **641** (FIG. 3A) has a spool **642** having equal-diameter lands a and b slidable in bore **643** and biased by biasing assembly **644** to the closed position shown. Control pressure line **564** is always connected to the bore **643** between lands a and b of spool **642** to a continuing portion of control pres-

sure line 564 and by restricted passage 646 to the chamber 647 at one end of bore 643 to act on land b of spool 642 to bias valve spool 642 toward regulating or open position when the force provided by regulated control pressure in chamber 647 overcomes the bias force of biasing assembly 644. In the regulating position, control pressure line 564 is connected to exhaust to the lower pressure mainline 112.

The biasing assembly 644 has a spring 648 seated in a vented spring chamber 649 and engaging annular thrust member 651 against the step 652 between spring chamber 649 and bore 643 in the closed position of spool 642 shown, and against land a of spool 642 in regulating position. The biasing assembly 644 also has 1st and 2nd pistons 653 and 654 slidable in series in piston bore 656 which has a closed end and opens into spring chamber 649 and is coaxial with valve bore 643.

High propulsion kidney pressure supplied by propulsion high kidney pressure line 629 to piston bore 656 biases 2nd piston 654 to engage land a of spool 642 in regulating position. Similarly, steer high kidney pressure line 657 is connected to the closed end of piston bore 656 to act on 1st piston 653 and through 2nd piston 654 on land a of spool 642. Thus the higher kidney pressure of the propulsion and steer high kidney pressures supplied by lines 629 and 657 determines the bias force. The control pressure in control pressure line 564 thus has an intermediate pressure value (e.g., 400 psi), determined by spring 648 which is higher than mainline pressure and increases in proportion to the higher of the propulsion or steer high kidney pressure to a maximum. Mainline 112 is connected by check valve 137 (FIG. 3A) for flow to control pressure line 564 to ensure that control pressure is never less than mainline pressure.

STEER CONTROL SYSTEM

Hydrostatic Transmission Controls

The hydrostatic drive controls 131 include steer hydrostatic drive controls 661 (FIG. 3D) to control the displacement position of steer pump ring 662 of variable displacement pump 90 of steer hydrostatic transmission 89. The fixed displacement motor 91 has a fixed ring (not shown). The steer pump ring 662 is pivotally mounted on a pin 663 fixed in housing 38 and positioned by 1st and 2nd steer servos 664 and 665 from null position (SN) for straight drive, and by 1st steer servo 664 to 1st position for left steer, and by 2nd steer servo 665 to 2nd position for right steer. Steer pump 90 and motor 91 have 1st and 2nd kidney passages 666 and 667, one passage having high kidney pressure and the other having low kidney pressure—depending on the direction of steer.

STEER CHARGING SYSTEM

Mainline 112 is connected (FIG. 3D) by 1st and 2nd check valves 668 and 669 respectively to 1st and 2nd steer kidney feed lines 671 and 672 to feed charging pressure to 1st and 2nd steer kidney passages 666 and 667.

STEER SHUTTLE VALVE

The steer shuttle valve 673 (FIG. 3D) is the same as propulsion shuttle valve 617 so like reference numerals (primed) have been used for the same parts, and reference is made to the above description of propulsion shuttle valve 617.

The 1st and 2nd steer kidney passages 666 and 667 are respectively connected by 1st and 2nd steer kidney pressure lines 674 and 675 to steer shuttle valve 673 which connects the high steer kidney pressure to steer high kidney pressure line 657 and low steer kidney pressure to steer low kidney pressure line 678.

2ND OR STEER CONTROL PRESSURE REGULATOR VALVE

The 2nd or steer control pressure regulator valve 139 (FIG. 3D) has a valve spool 682 having three equal-diameter lands a, b, and c, slidable in valve bore 683, and a larger diameter abutment portion 684 located within a spring 686 in a larger diameter piston bore 687. Piston 688 has a larger diameter than valve spool 682 and is slidable in piston bore 687 and biased to the right by spring 686. Exhaust 689 provides a vent between valve bore 683 and piston bore 687. Steer low kidney pressure line 678 is connected through a restriction 691 to chamber 692 at the closed end of piston bore 687 to act on and move piston 688 against the bias force of spring 686 to provide a net bias force acting on abutment portion 684 and thus on valve spool 682.

The steer control pressure lines 693 is connected to a closed chamber 694 to act on piston 696 which has a smaller diameter than valve spool 682, and is slidable in piston bore 697 and engages land a of spool 682 to provide a regulated steer control pressure bias force on valve spool 682 opposing the above net bias force. Exhaust 698 vents the step between valve bore 683 and piston bore 697.

When the low steer kidney pressure in chamber 692 provides a net bias force greater than the opposing regulated steer control pressure bias force, the 2nd or steer control pressure regulator valve 139 is in the connecting position shown, and control pressure line 564 is connected to steer control pressure line 693, and mainline 112 is connected to link feed line 699. When the regulated steer control pressure bias force overcomes the net bias force, the steer control pressure in line 693 is regulated at lower pressure values or is cut off, and link feed line 699 is connected to exhaust 701 to discontinue operation of steer override link 704 and steering as described below.

STEER SERVO VALVE

The steer servo valve 702 (FIG. 3D) is constructed like the above-described propulsion pump servo valve 563, so like reference numerals (double-primed) have been used and reference is made to the above description of valve 563.

In steer servo valve 702, valve spool 566″ is connected by lever or linkage 703 to steer override link 704. Valve sleeve 568″ is connected by linkage 706 to pin 707 on steer pump ring 662.

Steer control pressure line 693 supplies steer servo valve 702 which, like the above-described servo valves, functions in response to rotation of valve spool 566″ to selectively supply 1st and 2nd steer feed lines 708 and 709 which are normally connected through steer stroke limiter valve 592, respectively, to 1st and 2nd steer supply lines 711 and 712, and 1st and 2nd steer servos 664 and 665, to change steer pump displacement in the requested direction. This causes movement of steer pump ring 662 and follow-up movement through linkage 706 of valve sleeve 568″ to a closed position (as shown) to block flow to 1st and 2nd steer feed lines 708 and 709.

STEER STROKE LIMITER VALVE

The steer stroke limiter valve **592** (FIG. 3D) has a spool **714** having five equal-diameter lands a, b, c, d, and e, slidable in valve bore **715**. The 1st steer kidney pressure signal line **716** is connected to a 1st chamber **718** of the 1st biasing assembly **719** to act on the end of land e of spool **714**. The 1st biasing assembly **719** also has a 1st spring **721** seated in 1st chamber **718** and biasing 1st thrust washer **722** against a 1st shoulder **723** between valve bore **715** and 1st chamber **718** to contact land e of spool **714** to hold spool **714** in the central position shown.

The 2nd steer kidney pressure signal line **717** is connected to 2nd chamber **718'** of 2nd biasing assembly **719'** which, like 1st biasing assembly **719**, has a 2nd spring **721'**, 2nd thrust washer **722'**, and 2nd shoulder **723'**.

Opposing 1st and 2nd springs **721** and **721'** act on spool lands e and a to hold valve spool **714** in the central position shown as long as the pressure differential between the opposing 1st and 2nd steer kidney pressure signals acting on opposite lands e and a of spool **714** is insufficient to overcome the bias force of the opposing spring. The 1st and 2nd biasing assemblies **719** and **719'** have 1st and 2nd stop members **724** and **724'** to limit movement of valve spool **714**.

In the central position shown, 1st and 2nd steer feed lines **708** and **709** are connected by steer stroke limiter valve **592** respectively to 1st and 2nd feed branchlines **726** and **727** of 1st and 2nd steer supply lines **711** and **712**, to directly and freely connect 1st and 2nd steer feed lines **708** and **709** to 1st and 2nd steer supply lines **711** and **712**. The 1st and 2nd steer supply lines **711** and **712** also have respectively 1st and 2nd exhaust branchlines **728** and **729** blocked by spool lands e and a, with spool **714** in central position. Valve spool **714** also has 1st and 2nd passages **731** and **732** formed therein respectively connecting the space between spool lands a and b to 2nd chamber **718'** and connecting the space between spool lands d and e to the 1st chamber **718**.

When, for example, 1st steer kidney pressure excessively exceeds 2nd steer kidney pressure, or there is 1st steer kidney overpressure, this higher pressure in 1st chamber **718** overcomes 2nd spring **721'** and lower 2nd steer kidney pressure in 2nd chamber **718'** of the 2nd opposite biasing assembly **719'** to move valve spool **714** to restrict the flow and thus reduce the steer control pressure delivered from 1st steer feed line **708** to 1st feed branchline **726** and 1st steer supply line **711** to 1st steer servo **664** to permit displacement decrease to reduce 1st steer kidney pressure. Since the reaction force on steer pump ring **662** and thus steer control pressure requirements are proportional to high steer kidney pressure, reducing the steer control pressure will reduce high steer kidney pressure. If 1st steer kidney overpressure continues, valve spool **714** continues to move in the same direction and is connected by 1st steer kidney pressure signal line **716**, 1st chamber **718**, 2nd passage **732** in spool **714**, 2nd feed branchline **727**, and 2nd steer supply line **712**, to 2nd steer servo **665**; and 1st steer servo **664** is connected, via 1st steer supply line **711**, its 1st exhaust branchline **728** and 1st passage **731** in spool **714**, to 2nd chamber **718'** having low steer kidney pressure.

These connections provide a reversal of pressures in 1st and 2nd steer servos **664** and **665** to positively reduce displacement of pump **90**. Should there be 2nd steer kidney overpressure, the higher 2nd steer kidney signal pressure moves valve spool **714** in the opposite direction to reduce and then reverse control steer supply pressure in the same manner to reduce displacement to reduce high steer kidney pressure.

STEER SIGNAL VALVE

The steer signal valve **166** (FIG. 3D) has a spool **734** having equal-diameter lands a and b slidable in a bore **735**. Both spring **736** and steer signal pressure in 1st chamber **737**, supplied by steer signal line **326** from disconnect clutch valve **125**, act on the end of land b of spool **734** to provide a bias force to move spool **734** to the closed position shown. The input or 1st governor pressure line **165** is connected to a 2nd chamber **738** to provide a bias pressure to move valve spool **734** to the open position.

When steer signal valve **166** is in the closed position shown, link feed line **699** is blocked by land a of spool **734** and link supply line **741** is connected to exhaust **742** to disable steer override link **704**, as described below. In the open position, steer signal valve **166** connects link feed line **699** to link supply line **741** to supply and enable steer override link **704** to transmit steer request signals.

When manual range control **254** (FIG. 3B) is in neutral position, disconnect clutch valve **125** supplies mainline pressure, via steer signal line **326**, to 1st chamber **737**, so steer signal pressure and spring **736** close steer signal valve **166** against any engine speed governor pressure supplied by 1st governor line **165** to 2nd chamber **738** to exhaust link supply line **741** to disable steer override link **704** at all engine speeds.

When the manual range control **254** is shifted from neutral range to any operating range—forward, pivot steer, or reverse, the steer signal line **326** is exhausted at disconnect clutch valve **125** to exhaust 1st chamber **737**. Then engine governor pressure, at normal engine speeds providing sufficient power for steer operation supplied by 1st governor line **165** to 2nd chamber **738**, moves spool **734** against spring **736** to the open position. Then if the steer hydrostatic transmission **89** is properly supplied by steer hydrostatic charging system **135** so that there is a proper low kidney pressure in one of 1st and 2nd steer kidney passages **666** and **667**, the 2nd steer control pressure regulator valve **139** supplies mainline pressure to link feed line **699** which is connected by the open steer signal valve **166** to link supply line **741** to enable steer override link **704** to transmit steer request.

STEER OVERRIDE LINK

The steer override link **704** which is a follow-up hydromechanical link (FIG. 3D), has a spool **744** having two equal-diameter lands a and b, slidable in a sleeve bore **746** in sleeve **747** which is slidable in body bore **748** in fixed valve body **169**. The steer request lever **196** (FIG. 3C) is connected by steer request linkage **197'** to stem **750** of valve spool **744** to move spool **744** from the central null position (N) shown toward or to either 1st or 2nd steer position. The body bore **748** has 1st and 2nd end portions **751** and **751'** extending beyond the sleeve **747** at 1st and 2nd opposite ends **752** and **752'** of sleeve **747**. Body bore **748** also has at respectively opposite ends 1st and 2nd end walls **753** and **753'** enclosing 1st and 2nd spring chambers **754** and **754'** which are vented by 1st and 2nd exhausts **755** and **755'**. The 1st and 2nd springs **757** and **757'** are respectively located in 1st and 2nd spring chambers **754** and **754'** and seated on end walls **753** and **753'** and engage 1st and 2nd pistons **758** and **758'** which are slidable in 1st and 2nd bore end

portions 751 and 751' of body bore 748. Piston rod 759 is fixed to 1st piston 758, extends through 1st guide opening 761 in 1st end wall 753, and is operatively connected by lever or linkage 703 to rotate spool 566'' of steer servo valve 702 in response to reciprocal movement of 1st piston 758. Stem 750 of valve spool 744 extends through 2nd guide opening 761' in 2nd end wall 753', through, and is slidably sealed in opening 762 in 2nd piston 758'.

Link supply line 741 is connected through ports 740 which are axially extending grooves and ports in and between body and sleeve 747 for flow to sleeve bore 746 between lands a and b of spool 744 in all positions of spool 744 and sleeve 747. In response to null (N) or straight steer request, steer request linkage 197' positions spool 744 in the null position shown in FIG. 3D with lands a and b of spool 744 respectively blocking flow from sleeve bore 746 and between spool lands a and b to 1st and 2nd transfer passages 763 and 763' so 1st and 2nd springs 757 and 757', center sleeve 747, and 1st and 2nd pistons 758 and 758' abut 1st and 2nd ends 752 and 752' of sleeve 747. Thus piston rod 759 and steer servo valve 702 are in null (N) position. The 1st transfer passage 763 has a port 764 blocked by land a of spool 744 in null position and a groove 765 in sleeve 747 connected by port 766 to sleeve bore 746 between spool land a and piston 758 to 1st end portion 751 of body bore 748. 2nd transfer passage 763' is similar so the same reference numerals (primed) are used.

On steer request toward or to 1st steer position, steer request linkage 197' moves valve spool 744 upwardly in FIG. 3D to connect link supply line 741 via sleeve bore 746 and 1st transfer passage 763 to 1st end portion 751 of body bore 748 to act on 1st piston 758 to move 1st piston 758 toward 1st steer position. During this follow-up movement, the leakage past 2nd piston 758' is larger than the leakage past spool land b so leakage fluid therebetween does not affect follow-up movement. The 2nd spring 757' acting through 2nd piston 758' engaging 2nd end portion 752' of sleeve 747 causes sleeve 747 to follow 1st piston 758, and when 1st end portion 752 engages 1st piston 758, the spool lands a and b will again block both 1st and 2nd transfer passages 763, 763' when the position toward or to 1st position of 1st piston rod 759 corresponds with such position of valve spool 744.

Steer request toward or to 2nd steer position moves valve spool 744 in the opposite direction similarly connecting link supply line 741 past land b of spool 744 to 2nd transfer passage 763' to move 2nd piston 758' against 2nd spring 757'. The piston rod 759, 1st piston 758, and sleeve 747 similarly follow 2nd piston 758', and in the requested steer position spool land b blocks flow to 2nd transfer passage 763' to stop movement.

VEHICLE SERVICE BRAKE SYSTEM

The vehicle service brake system 141 has a brake apply valve 771, a brake signal valve 772, and a brake coolant valve 773.

BRAKE APPLY VALVE

The brake apply valve 771 (FIG. 3E) has a regulating spool 776 having a small diameter land a slidable in a small diameter bore 777, and a larger diameter land b slidable in a large diameter bore 778, and a control spool 779 of the same large diameter as spool land b, and having equal-diameter lands c and d slidable in large diameter bore 778. Spring 781 is seated in recess 782 formed in control spool 779 at the end facing regulating spool 776 and engages large diameter land b of regulating spool 776. A rod 783 is secured to control spool 779 at the base of recess 782 and extends through rod bore 784 in regulating spool 776 and has a stop 786 secured on the end of rod 783 to limit relative separating movement of regulating and control spools 776 and 779 in the brake-OFF position shown. Rod 783 has sufficient clearance in rod bore 784 for free movement of regulating spool 776 on rod 783 and for free flow of leakage fluid through rod bore 784. The large diameter bore 778 between regulating and control spools 776 and 779 is vented to exhaust 787. Chamber 788 at the end of small diameter bore 777 may be vented by its exhaust 789 and/or, via rod bore 784, to exhaust 787.

As the vehicle brake pedal 791 is moved for increasing brake demand, a mechanical or hydraulic linkage 792 applies a force to the end 793 of control spool 779 to move control spool 779 to compress spring 781 to apply a proportionately increasing apply force to large land b of regulating spool 776. This apply force moves regulating spool 776 from brake-OFF position, connecting brake apply line 794 to exhaust 796 to brake-ON regulating position connecting mainline 112 or exhaust to regulate brake apply pressure proportionally to this apply force and brake demand. The control spool 779 in brake-OFF position connects brake-ON signal line 797 to exhaust 798 and then in response to initial movement of pedal 791 for low brake apply or during the slack take-up movement, blocks exhaust 798 and connects mainline 112 to brake-ON signal line 797 to provide brake-ON signal pressure. Brake apply line 794 has a restricted exhaust 799 to provide flow to stabilize regulation of brake apply pressure.

BRAKE SIGNAL VALVE

The brake signal valve 772 (FIG. 3E) has a spool 802 having equal-diameter lands a and b slidable in valve bore 803. Piston 804, having a stem 805, is slidable in larger piston bore 806 which at one end is closed by end wall 807 and at its other end is coaxially open to valve bore 803 at the step 808 between these valve and piston bores. A spring 809 is seated between piston 804 and annular member 811 seated on step 808 and biases piston 804 against end wall 807 in 2–3 ratio range and brake-OFF position shown.

The 1st apply line 239 is connected by restriction 812 to 1st chamber 813 between end wall 807 and piston 804. Piston bore 806 is vented at step 808 by an exhaust 814. The brake-ON signal line 797 is connected to brake chamber 816 at the closed end of valve bore 803 so brake-ON signal pressure acts on land b of spool 802.

In 1st ratio range, 1st apply pressure in 1st chamber 813 is fed through the restriction 812 and thus slowly moves piston 804 against the spring 809 so piston stem 805 engages spool land a to position valve spool 802 in brake signal-OFF position shown, blocking brake apply line 794 and connecting brake apply signal line 184 to exhaust 817 so there is no brake apply signal pressure supplied to brake apply signal line 184. In 2nd and 3rd ratio ranges, 1st apply pressure is exhausted via restriction 812 so spring 809 slowly moves piston 804 to the released position shown, so valve spool 802 is free to move to either the brake apply signal-ON or OFF position.

When the brakes are off, since there is no brake apply pressure in brake apply line 794 and no brake-ON signal pressure in brake signal line 797, valve spool 802 connects brake apply signal line 184, if in brake-OFF posi-

tion, to exhaust 817 and, if in the brake-ON position, to exhausted brake apply line 794, so there is no brake apply signal pressure. When the brakes are applied before any brake apply or during the slack take-up period, brake-ON signal pressure via brake-ON signal line 797 is supplied to brake chamber 816 to move valve spool 802 to the brake-ON position connecting brake apply line 794 to brake apply signal line 184 to supply brake apply pressure to ratio governor 162.

BRAKE COOLANT VALVE

The brake coolant valve 773 (FIG. 3F) has a spool 822 having one large diameter land a slidable in a large diameter bore 823. Valve spool 822 has at one end thereof a small diameter guide stem 824 slidably mounted in a guide bore 826 in the end wall 827 of sleeve member 828. Sleeve member 828 fits and is fixed and sealed in large diameter valve bore 823 to body 169. A spring 829 is seated on end wall 827 and engages valve spool land a to bias spool 822 so that land a is sealed against the step 831 between large diameter valve bore 823 and the small diameter bore 832 in the brake-OFF position as shown. Valve spool 822 also has at the other end thereof a rod 833 with a piston 834 slidably in small diameter bore 832. Brake-ON signal line 797 is connected to a chamber 836 at the closed end of small diameter bore 832 to act on piston 834 to move valve spool 822 to the brake-ON position.

Lube line 115, which also feeds the hydro-lube system 837 for lubrication of both the propulsion and steer hydrostatic transmissions 39 and 89, is connected to a port 838. Port 838 is an annular recess in the large diameter valve bore 823 and is wider than land a of spool 822 so the port 838 is not blocked in any position of land a. In the brake-OFF position shown, lube line 115 is connected via port 838 to large diameter valve bore 823 to supply gear lube line 839 and gear lube system 840 for lubrication of all gearing of the propulsion and steer drives 13 and 14. Such lube systems have a downstream restriction and then exhaust fluid to the sump 111, so gear lube line 839 has a lube relief valve 841.

In this brake-OFF position, one side of land a of spool 822 is sealed against step 831 so there is no lube fluid flow through brake coolant valve 773 directly to brake coolant line 842, but gear lube line 839 supplies a metered amount of lube fluid via restricted passage 844 to brake coolant line 842 and brake cooling system 843.

On initial brake demand, brake apply valve 771 supplies brake-ON signal line 797 which is connected to chamber 836 of brake coolant valve 773 to move valve spool 822 against the spring 829 to the brake-ON position where the other side of land a of spool 822 engages the end 846 of sleeve member 828 which functions like step 831 so lube line 115 is disconnected from gear lube line 839 and connected via port 838 to brake coolant line 842 and brake cooling system 843 for high volume coolant flow just before the vehicle service brakes 37 and 37' are applied. The brake coolant line 842 and cooling system 843 have some downstream restriction so brake coolant pressure in brake coolant line 842 supplies, via restricted passage 844, a metered amount of lube fluid to gear lube line 839 and system 840.

OPERATION

Fluid Supply

While the above description describes the operation of the controls with reference to each component, it is believed desirable to describe the overall drive operation.

When the input shaft 18, or another suitable shaft in input drive 12, is driven by engine or overrun drive, the main pump 110 (FIG. 3F) is driven and supplies mainline pressure to mainline 112 at a constant pressure value (e.g., 150 psi) when flow requirements are met.

Mainline 112 supplies the following parts of control system 11: basic signal controls 116 for automatic operation; the valves controlled electrohydraulically by the manual range control 254; disconnect clutch valve 125; neutral shift valve 127; forward and reverse signal valves 128 and 129; inhibitor valve 126; the propulsion and steer charging systems 134 and 135 for propulsion and steer hydrostatic transmissions 39 and 89; the similarly driven hydrostatic control pump 136; zero stroke indicator valve 132; the 2nd or steer control pressure regulator valve 139 to supply link feed line 699 for steering; and vehicle service brake system 141. Hydrostatic control pump 136 supplies 1st propulsion and 2nd steer control pressure lines 564 and 693. Main regulator valve 114 supplies overage fluid flow to lube line 115 for the lube systems.

Mainline 112 supplies basic signal controls 116. Modulator valve 117 supplies modulator pressure directly proportional to compressor discharge pressure, and thus engine power, to engine power signal line 146. Output speed governor 118 and output governor signal valve 119 supply output or 2nd governor pressure proportional to output speed above a minimum speed to output or 2nd governor signal pressure line 157. Engine or input governor 161 supplies 1st or input governor pressure to 1st or input governor pressure line 165. The advance return signal system 160, having ratio governor 162, feeds advance and return signals in response respectively to overspeed or underspeed relative to a controlled set speed via return demand control 206 and rate of ratio change control 203 to propulsion ratio control 121. In ratio governor 162, the flyweight assembly 172 and spring 173 determine idle set speed for idle throttle, no steer request, idle engine power, and no brake demand. The throttle and steer request device 190, through throttle demand and steer request cams 188 and 189, increase the set speed in predetermined proportions to a higher set speed, due either to the throttle (torque demand) or steer request. The power request device 185 is supplied by power signal line 183 which is supplied alternatively by the engine power signal line 146 and vehicle apply signal line 184 to increase the set speed during drive operation in proportion to increasing compressor discharge pressure or engine power and, during vehicle brake operation, in proportion to brake demand or brake torque. The rate of ratio change control 203 provides fast increase and slow decrease of return signal pressure, and slow increase in 2nd and 3rd ratio range and slower increase in 1st ratio range, and fast decrease of advance signal pressure for improved ratio change control stability.

Thus the complete cross drive train 10 and control system 11 are supplied with pressure fluid and ready for operation in response to manual selection of a manual range, throttle, steer, engine power or brake demand, and engine speed and output speed.

MANUAL RANGES

Neutral

On initial start-up of the engine and thus the initial supply of mainline pressure, ratio governor **162** is biased by spring **173** to supply only return signal pressure, so the displacement of pump **41** is substantially zero for a smooth unloaded start-up. When the engine is started, the vehicle normally is stationary and gear ratio control **420** disengages all ratio ranges. Also, manual range control (FIG. 3B) **254** is in the neutral (N) position to energize solenoid valves A, C, F, and G (see TABLE I), to position the valves in the hydraulic range controls **255** for neutral operation. Disconnect clutch valve **125** is in the neutral (N) or disconnect position shown connecting mainline **112** to steer signal line **326** to control steer signal valve **166** to prevent steer operation and to exhaust and disestablish disconnect clutch **24**, so there is no input drive to propulsion and steer drives **13** and **14**. Neutral shift valve **127** is in the neutral position shown, so advance line **204** is connected to neutral advance line **264**. Both neutral advance line **264** and return line **205** are connected, via neutral shuttle valve **330** and 1-2-3 return line **331**, to return signal control valve **332** which is in neutral low (1st) position to continuously connect a return signal, via 1st or low return line **339**, to zero stroke bias valve **363**. The return signal control valve **332** exhausts the 2-3 return line **341**, so there is no supply to advance signal control valve **344** and 1st (plus) and 2nd (minus) ratio change feed lines **361** and **358**.

The neutral shift valve **127** exhausts the range feed line **267** so 1st brake **65**, 2nd clutch **88**, and 3rd clutch **72**, all ratio-establishing devices, are disestablished for positive neutral. Mainline **112** is connected via inhibitor valve **126** to 3rd chamber feed line **285** and reverse chambers **296** and **296'** of forward and reverse signal valves **128** and **129**.

With solenoid valves C and F energized, forward signal valve **128** is in forward position shown and reverse signal valve **129** is in reverse position, so both forward and reverse signal lines **288** and **289** are exhausted. Thus, the 1-2-1 and 2-3-2 shift signal valves **396** and **386** are not supplied and not functioning. The 1st or low return line **339** supplies zero stroke bias valve **363**, which is biased by spring **374** to the reverse position shown in FIG. 3A, since there is no forward signal in forward signal line **288** in neutral manual range. The zero stroke bias valve **363**, whose spool **364** is positioned by propulsion pump ring **370**, selectively controls the supply from 1st or low return line **339** to reverse return line **383** (a minus signal) and to 1st forward return line **384** (a plus signal). These lines are respectively connected by 2nd or minus and 1st or plus shuttle valves **562** and **561** to 2nd or minus and 1st or plus actuator lines **558** and **557** to position ratio change actuator **543** and, by linkage **567**, place pump servo valve **563** in a reverse bias position so the propulsion hydrostatic drive controls **133**, by selective supply of control pressure to 1st (plus) servo motor **537** and 2nd (minus) servo motor **538** for regulating operation, to position pump ring **370** in reverse bias (RB) position (FIGS. 3A and 4).

In neutral, the overpressure system **573**, having propulsion stroke limiter valve **588** which is open as there is no load, and high kidney overpressure and safety mode valve **591** which is open as gear ratio control **420** is not in 2nd or 3rd ratio range, merely connect pump servo valve **563** via 1st (plus) and 2nd (minus) pump control feed lines **571** and **572**, 1st (plus) and 2nd (minus) pump control connecting lines **589** and **590**, and 1st (plus) and 2nd (minus) pump control supply lines **574** and **575**, to 1st (plus) and 2nd (minus) servo motors **537** and **538**. With the ratio change actuator **543** in reverse bias position (RB), the connection by linkages **567** and **577** positions motor servo valve **576** to connect control pressure line **564** to 2nd (or minus) motor control supply line **580** and 2nd (minus) motor servo **540** to move motor ring **401** to the maximum negative (M−) position (FIGS. 3A and 4), a limit position, with 1st (plus) motor servo **539** having its piston engaging its cylinder end wall, as shown in FIG. 3A.

PIVOT STEER

When manual range control **254** is moved from neutral (N) position to pivot steer (PS) position, as in neutral, solenoid valves A, C, and F, remain energized so neutral shift valve **127** and forward and reverse signal valves **128** and **129** are in the same positions as described above for Neutral. Also, since 2nd and 3rd ratio ranges are not engaged, inhibitor valve **126** is in neutral-1st position, and functions as in neutral position. However, solenoid valve G is deenergized and solenoid valve H is energized to shift disconnect clutch valve **125** to the connect position, connecting mainline **112** to disconnect clutch **24** to establish input drive to the propulsion and steer drives **13** and **14** and to exhaust steer signal line **326**. When the low steer kidney pressure in one of 1st (**666**) and 2nd (**667**) steer kidney passages (FIG. 3D) has proper pressure values, steer control pressure regulator valve **139** connects propulsion control pressure line **564** to steer control pressure line **693** and mainline **112** to the link feed line **699**. When the engine is operating in a proper speed range, input governor pressure supplied by 1st or input governor pressure line **165** shifts steer signal valve **166**, moving spool **734** against spring **736** to connect link feed line **699** to link supply line **741**, so steer override link **704** transmits manual steer demand or request from steer request lever **196** to steer servo valve **702**. In response to 1st and 2nd steer request, steer servo valve **702** connects steer control pressure line **693** respectively to 1st and 2nd steer feed lines **708** and **709** and 1st and 2nd steer servos **664** and **665**, to move steer pump ring **662** to or toward 1st and 2nd maximum displacement positions to drive the steer drive **14** in 1st and 2nd, or in opposite directions for steering in one or the other direction.

In pivot steer manual range, zero stroke bias valve **363** functions the same as in neutral manual range to place pump ring **370** in reverse bias displacement (RB) position and motor ring **401** in maximum negative displacement (M−) position (FIGS. 3A and 4).

Since range feed line **267** (as in Neutral), is exhausted at neutral shift valve **127** (FIG. 3B), no ratio range drive-establishing devices are engaged, so there is no propulsion drive.

FORWARD

On shifting manual range control **254** to forward position, solenoid valves B, C, E, and H are energized. Thus disconnect clutch valve **125**, as in *Pivot Steer*, engages disconnect clutch **24** and steer signal line **326** is exhausted, so there will be normal steer operation at proper low kidney passage pressure and proper engine speed, as in pivot steer. Since 2nd and 3rd ratio ranges are not engaged at idle throttle, inhibitor valve **126** (as

in Neutral and Pivot Steer), is in the normal position shown in FIG. 3B, connecting mainline **112** to 3rd chamber feed line **285** for control of forward and reverse signal valves **128** and **129**. Since solenoid valve A is deenergized and solenoid valve B is energized, neutral shift valve **127** moves to forward-reverse position, so mainline **112** is connected to range feed line **267**, advance line **204** is connected to automatic advance line **266**, and neutral advance line **264** is exhausted.

Both the forward and reverse signal valves **128** and **129** are in forward position, as shown, to connect mainline **112** to forward signal line **288** and to exhaust reverse signal line **289**. Return line **205** is connected by neutral shuttle valve **330** to 1-2-3 return line **331** to feed return signal control valve **332** which in neutral, pivot steer, and 1st ratio range of forward manual range, supplies the return signal pressure to 1st or low return line **339**, and in 2nd and 3rd ratio ranges, to 2-3 return line **341**. The forward signal is supplied by forward signal line **288** to move valve sleeve **366** of zero stroke bias valve **363** to forward position. At idle throttle, there is no advance signal pressure and only a return signal pressure supplied by ratio governor **162**, via return signal control valve **332** and 1st or low return line **339** to zero stroke bias valve **363**.

The zero stroke bias valve **363**, in forward manual, and idle throttle, selectively regulates supply of return signal pressure to reverse return line **383** (a minus signal) and to 1st forward return line **384** (a plus signal), to control ratio change actuator **543**. The ratio change actuator **543** controls the position of pump and motor servo valves **563** and **576** to regulate the position of pump ring **370** at forward bias (FB) position FIGS. **3A** and **4**), and to place motor ring **401** at maximum negative (M−) position (FIGS. **3A** and **4**) which, since 1st brake **65** is engaged, is a vehicle creep drive (e.g., ½ to 1 mph).

On shifting manual range control **254** to forward position, neutral shift valve **127** shifts to drive position and supplies mainline **112** pressure via range feed line **267**; 2-3 overpressure cut-off valve **431** which is always open when 2nd and 3rd ratio drives are not engaged; 1st range supply line **442**; priority valve **456**; 3rd (after priority) range supply line **458**; 1-2 shift valve **491** is 1st position, and 1st apply line **239** to 1st brake **65**, to engage 1st ratio range. The 1-2 shift valve **491** is biased by spring **497** to 1st position shown in FIG. **3E** as 1-2 shift signal line **489** is exhausted by ratio shift signal valve **257** when positioned by the return signal supplied by return line **205**, directly to exhaust **485** or, when positioned by the delayed automatic advance signal supplied by line **423** via the 1-2-1 shift signal line **402** and 1-2-1 shift signal valve **396**, to exhaust **403**.

In order to simplify the following description of the overall speed ratio change operation in forward range, it is assumed that there is continuous overspeed, engine speed is greater than set speed, so ratio governor **162** continuously provides an advance signal to increase overall speed ratio from "creep" to maximum speed ratio and then the opposite, under speed and a return signal, to decrease overall speed ratio from a maximum to creep speed ratio. However, it should be noted that ratio governor **162** selectively supplies advance or return signals to increase or decrease and thus regulate the overall speed ratio so engine speed equals or matches the changing set speed so advance or return signals are provided at any engine speed in response to overspeed or underspeed to regulate engine speed to match the set speed.

When forward range idle throttle operation is established, assuming the throttle is increased at a sufficient rate and amount, such as full for accelerating, the vehicle under such load conditions that there is continuous overspeed, ratio governor **162** will continuously provide an advance signal to advance feed line **201**. This advance signal, in the absence of a return demand signal in return demand signal line **227**, is connected by advance return signal system **160**, advance control cut-off valve **122**, and rate of ratio change control having restriction and check valve device **233**, and 1st range advance control valve **123**, operating in 1st ratio range to further delay the increase of the advance signal pressure feed to advance line **204**. This advance signal pressure in advance line **204** is connected by neutral shift valve **127**, via automatic advance line **266**, for connection by advance signal control valve **344** in 1-3 position shown (FIG. **3B**), to 1st (plus) ratio change feed line **361**, for connection by 1st (plus) shuttle valve **561** to actuator **543** to move or advance piston **544** from forward bias (FB) position to a 1-3 position for increasing positive pump displacement, and then to a 1-3 shift position (1-3S) for decreasing negative motor displacement.

Actuator **543**, through linkages **567** and **577**, moves pump and motor servo valves **563** and **576**. Pump servo valve **563**, as its spool **566** is moved from forward bias (FB) position to 1-3 position, connects propulsion control pressure line **564** to 1st (plus) pump control feed line **571**, connected via overpressure system **573** and 1st (plus) pump control supply line **574**, to 1st pump servo **537** to move pump ring **370** from forward bias (FB) position (FIGS. **3A** and **4**) to maximum positive pump displacement position (P+) at a 1st ratio transition speed (1T) where 2nd pump servo **538**, due to its piston engaging the cylinder end wall, limits or prevents further increase of pump displacement due to movement of pump servo valve spool **566** from 1-3 transition position (1-3) to 1-3 shift position (1-3S).

Motor servo valve **576**, as its spool **566'** moves from forward bias (FB) position to 1-3 position (1-3), continuously supplies control pressure from 1st propulsion control pressure line **564**, via 2nd motor control supply line **580**, to 2nd motor servo **540** (a minus servo) to position motor ring **401** in maximum negative (M−) position (a limit position) to provide maximum negative motor displacement (M−). The 1st motor servo **539** acts as a limit stop. Then on movement of motor servo valve spool **566'** from 1-3 position (1-3) to 1-3 shift position (1-3S), connects 2nd motor control supply line **580** to exhaust and supplies control pressure to 1st motor control supply line **579** and 1st motor servo **539** (a plus servo) to move motor ring **401** to an intermediate negative displacement position (MI−), FIGS. **3A** and **4**). In 1st ratio range drive, this continuous displacement change, first increasing positive pump displacement and then decreasing negative motor displacement, continuously increases overall speed ratio. At the 1-2-1 shift point, the gearing drives the input and output members of 2nd clutch **88** at the same speed for a synchronous shift.

At the 1-2-1 shift point and speed, motor ring **401** positions 1-2-1 shift signal valve **396** to supply a 1-2-1 shift signal, via 1-2-1 shift signal line **402**, to ratio shift signal valve **257**. Since delayed automatic advance signal line **423** supplies a signal to position ratio shift signal

valve 257 in upshift position, the 1-2-1 shift signal line 402 is connected to 1-2 shift signal line 489 which acts on 3rd piston 477 of ratio shift signal valve 257 to provide a hysteresis force on ratio shift signal valve spool 468 to prevent hunting, and acts on 1-2 shift valve 491 to effect a 1-2 shift. The 1-2 shift valve 491 in 2nd position connects 1st apply line 239 to 1st exhaust line 503 and 3rd range supply line 458 via 2-3 feed line 283, and 2-3 shift valve 492 in 2nd ratio position to 2nd clutch line 356 and 2nd clutch 88 to disengage 1st brake 65 and engage 2nd clutch 88 for 2nd ratio drive. The 1-2 shift valve 491, in 2nd position, also connects 2nd range supply line 457 to 2nd upshift chamber 502 to hold 1-2 shift valve 491 in 2nd position as 1-2-1 shift signal in line 402 and 1-2 shift signal in line 489 will be exhausted as soon as motor displacement is reduced in 2nd ratio range drive.

The 2-3 shift feed line 283 is also connected to and moves return signal control valve 332 to 2-3 position (FIG. 3B), inhibitor valve 126 to 2-3 position and safety mode valve 591 to safety mode or 2-3 position. The 2nd clutch apply line 356 is connected to and moves advance signal control valve 344 to 2nd reverse position. Return signal control valve 332, in 2-3 position, connects 1-2-3 return line 331 via 2-3 return line 341 to advance signal control valve 344 which in 2nd reverse position connects this return signal to 1st (plus) ratio change feed line 361 and automatic advance line 266 to 2nd (minus) ratio change feed line 358. These 1st and 2nd ratio change feed lines 361 and 358, connected respectively via 1st (or plus) and 2nd (or minus) shuttle valves 561 and 562 to ratio change actuator 543, are reversed relative to operation in 1st ratio range so the advance signal is now a minus signal moving actuator 543 in a negative direction to 2nd position (2), a limit position, engaging stop member 553 of stop piston 551, held in 2nd stop position by forward signal pressure supplied by forward signal line 288. As actuator 543 moves from 1st shift position (1-3S), FIG. 3B, toward 2nd position (2), the similarly moving motor servo valve 576 connects control pressure line 564 to 2nd motor servo 540 to move motor ring 401 to increase negative motor displacement from intermediate negative position (MI−) to maximum negative limit position (M−), and the similarly moving pump servo valve 563 supplies control pressure to 1st pump servo 537 to hold pump ring 370 in maximum positive displacement position (P+). Then, with further movement of ratio change actuator 543 and pump and motor servo valves 563 and 576, the pump servo valve 563 supplies control pressure to 2nd pump servo 538 to move pump ring 370 to decrease pump displacement to a 2nd speed intermediate negative displacement position (P2−), and motor servo valve 576 continues to supply control pressure to 2nd motor servo 540 to hold motor ring 401 in maximum negative position (M−). This displacement change provides continuous overall speed ratio increase in 2nd ratio range drive.

At the 2-3-2 shift point and speed, the pump ring 370, in 2nd speed intermediate displacement position (P2−), controls 2-3-2 shift signal valve 386 to supply 2-3-2 shift signal line 391 which is connected by the ratio shift signal valve 257, still held in upshift position by delayed automatic advance signal pressure supplied by line 423, to 2-3 shift signal line 487. The 2-3 shift signal line 487 is connected so that 2-3 shift signal pressure acts on 4th piston 478 of ratio shift signal valve 257 to prevent hunting of ratio shift signal valve 257 and to act on spool 493' to upshift 2-3 shift valve 492 to 3rd position. The 2-3 shift valve 492, in 3rd position, connects 2-3 feed line 283 to 3rd apply line 512 to engage 3rd clutch 72 for 3rd ratio range and to act on piston 494' to hold 2-3 shift valve 492 in 3rd position, and connects 2nd clutch apply line 356 to 2nd exhaust line 504 to disengage 2nd clutch 88 for a synchronous 2-3 shift.

Return signal control valve 332, inhibitor valve 126, and safety mode valve 591 remain in 2-3 position but on exhaust of 2nd clutch apply line 356, advance signal control valve 344 returns to 1-3 position so automatic advance line 266 and 2-3 return line 341 are connected to ratio change actuator 543 as in 1st ratio range to similarly increase displacement of pump 41 to the positive maximum limit position (P+) while holding displacement of motor 42 at the maximum negative limit position (M−) at 3rd transition speed (3rd T) and then decrease negative motor displacement from the maximum negative limit position (M−) to an intermediate negative position (MI−) to continuously increase overall speed ratio to the maximum value.

When the throttle 194 is returned to Idle or at least sufficiently so there is a constant engine underspeed, only a return signal is provided by ratio governor 162. This return signal is supplied without delay by the rate of ratio change control 203 via return line 205, neutral shuttle valve 330, return signal control valve 332 in 2-3 position, 2-3 return line 341 to advance signal control valve 344. Advance signal control valve 344, in 1-3 position, connects 2-3 return line 341 to 2nd (or minus) ratio change feed line 358 and via 2nd (or minus) shuttle valve 562 to move actuator 543 toward 2nd position (2). Actuator 543 similarly moves pump and motor servo valves 563 and 576 to return pump and motor rings 370 and 401 and thus change pump and motor displacement in the same manner on the same curves (FIG. 4), so pump displacement is at the intermediate negative 2nd speed value (P2−), and motor displacement is at maximum negative limit value (M−) described above at the 2-3-2 shift position.

Then 2-3-2 shift signal valve 386 provides a 2-3-2 shift signal via 2-3-2 shift signal line 391 to ratio shift signal valve 257, which is now placed in downshift position by the return signal supplied by return line 205 and connects 2-3-2 shift signal line 391 to 3-2 shift signal line 486. The 3-2 shift signal line 486 is connected to act on 1st piston 473 of ratio shift signal valve 257 to prevent hunting of this valve and to downshift chamber 498' of 2-3 shift valve 492 so there are balanced pressure forces on spool 493', and spring 497' downshifts 2-3 shift valve 492 to 2nd position. The 2-3 shift valve 492, in 2nd position, exhausts 3rd apply line 512 to 3rd exhaust line 513 and connects 2-3 feed line 283 to 2nd clutch apply line 356 for a 3-2 ratio shift. Since as hydrostatic ratio is decreased in 2nd ratio range, 3rd apply line 512 is exhausted and 3-2 shift signal line 486, via ratio shift signal valve 257, 2-3-2 shift signal line 391, and 2-3-2 shift signal valve 386 is exhausted, there are no signal forces on spool 493', so spring 497' holds spool 493' of the 2-3 shift valve 492 in 2nd position.

The 2nd clutch apply line 356 supplies the 2nd apply pressure to shift the advance signal control valve 344 to 2nd position to provide the above-described reversed connection, so 2-3 return line 341 is connected to 1st (plus) ratio change feed line 361 to move actuator 543 toward 1-3 position and similarly move pump and motor servo valves 563 and 576. The pump servo valve 563 connects 1st propulsion control pressure line 564 to

1st (plus) pump control feed line **571** to supply, via overpressure system **573**, 1st pump servo **537** to move pump ring **370** from the 2nd maximum negative displacement position (P2−) to maximum positive displacement position (P+), a limit position maintained with continued plus directional movement of pump servo valve **563**. The motor servo valve **576** connects control pressure line **564** first, via 2nd (minus) motor control supply line **580** to 2nd motor servo **540** to hold motor ring **401** in maximum negative limit position (M−), and then when pump ring **370** reaches maximum positive displacement position (P+), to supply, via 1st (plus) motor control supply line **579** to 1st motor servo **539**, to move motor ring **401** from maximum negative displacement limit position (M−) to motor intermediate negative displacement position (MI−) at the 1-2-1 shift point.

At the 1-2-1 shift point, ring **401** positions 1-2-1 shift valve **396** to supply 1-2-1 shift signal line **402**, to ratio shift signal valve **257**, which is held in the downshift position by return pressure supplied by return line **205**, and connects 1-2-1 shift signal line **402** to 2-1 shift signal line **488**. The 2-1 shift signal line **488** is connected, via downshifted 2-3 shift valve **492** and 2-1 shift signal transfer line **514**, to downshift chamber **498** of 1-2 shift valve **491**. The pressure in downshift chamber **498** and spring **497** downshifts 1-2 shift valve **491** against the hold-pressure in 2nd upshift chamber **502**, so 2-3 feed line **283** is exhausted via 2nd exhaust line **504** and 3rd range supply line **458** is connected to 1st apply line **239** to engage 1st brake **65** for 1st ratio range drive. The exhausting of 2-3 feed line **283**, since it is connected by downshifted 2-3 shift valve **492** to 2nd apply line **356**, disengages 2nd clutch **88**.

Also, on exhaust of 2-3 feed line **283**, inhibitor valve **126** returns to 1st or normal position (1-3); safety mode valve **591** returns to inactive position; and return signal control valve **332** returns to 1st position. Then, assuming that only a return signal is supplied by ratio governor **162** to return line **205** via neutral shuttle valve **330**, return signal control valve **332** and 1st or low return line **339** to zero stroke bias valve **363**, at all displacement positions of pump ring **370** above the forward bias position (FB), zero stroke bias valve **363** connects 1st or low return line **339** to reverse return line **383** (a minus line) which is connected by 2nd (or minus) shuttle valve **562** and 2nd actuator line **558** to return actuator **543** to forward bias position (FB). Actuator **543** controls pump and motor servo valves **563** and **576** to control displacement of pump **41** and motor **42** to decrease overall speed ratio in a manner opposite to the above-described control by the advance signal to increase overall speed ratio in 1st ratio range. During return movement, at first pump servo valve **563** holds pump ring **370** at maximum positive displacement position (P+) as motor servo valve **576** returns motor ring **401** from intermediate negative displacement position (MI−) to maximum negative limit position (M−). Then pump servo valve **563** returns pump ring **370** from maximum positive displacement position (P+) to forward bias position (FB) as motor servo valve **576** holds motor ring **401** in maximum negative limit position (M−).

In 2nd and 3rd ratio ranges, 2-3 feed line **283** supplies a 2-3 feed pressure to move inhibitor valve **126** to 2-3 position and thus exhaust 3rd chamber feed line **285** so pressure is not supplied to reverse pressure chambers **296** and **296'**, respectively, of forward and reverse signal valves **128** and **129**. Thus manual range control **254** is rendered ineffective to shift forward and reverse signal valves **128** and **129** out of forward position in 2nd and 3rd ratio ranges.

The 2-3 feed in line **283** also moves safety mode valve **591** to 2-3 safety mode position (FIG. 3A), so there is overpressure control of plus or minus control pressure supplied by 1st and 2nd pump control supply lines **574** and **575** to 1st and 2nd pump servos **537** and **538**, not only by propulsion stroke limiter valve **588** as in 1st and reverse ratio ranges, but also by safety signal valve **451** and 2-3 overpressure cut-off valve **431**. Safety mode valve **591**, in 2-3 position, as in 1st position, continues the connection of plus and minus control pressures to 1st and 2nd pump servos **537** and **538** and also connects these control pressures to safety signal valve **451** which, in response to overpressure and only in 2nd and 3rd ratio ranges, provides an overpressure signal in high pressure signal line **452**. The 2-3 overpressure cut-off valve **431**, in response to the overpressure signal in high pressure signal line **452**, cuts off the connection of range feed line **267** to 1st range supply line **442** so there is no pressure supplied to the 1-2 and 2-3 shift valves **491** and **492**, so 1st, 2nd, and 3rd ratio ranges are disengaged and there is no propulsion drive.

The lock system **449** is then effective to hold 2-3 overpressure cut-off valve **431** in cut-off position (FIG. 3F), until output speed is reduced to a low speed and pump ring **370** is substantially at zero stroke or displacement position for reestablishing 1st ratio drive under low load.

REVERSE RANGE

On shifting manual range control **254** to reverse range, both forward and reverse signal valves **128** and **129** are moved to reverse position to supply reverse signal line **289** and to exhaust forward signal line **288**. Disconnect clutch valve **125** is in connect position, as in forward range, engaging disconnect clutch **24** and exhausting steer signal line **326**, so steer drive **14** may be operated. Neutral shift valve **127** is in the same forward-reverse position connecting advance line **204** to automatic advance line **266**.

Return line **205** is connected, as in 1st ratio range, via neutral shuttle valve **330**, return signal control valve **332** in normal position, and 1st or low return line **339**, to supply zero stroke bias valve **363**. In the absence of an advance signal from ratio governor **162**, at idle throttle and with valve sleeve **366** of zero stroke bias valve **363** in reverse position, since forward signal line **288** is exhausted, zero stroke bias valve **363** regulates the supply to reverse return line **383** or 1st forward return line **384** to position pump ring **370** in reverse bias position (RB), as in neutral manual range for reverse creep drive. Reverse signal line **289** moves advance signal control valve **344** to 2nd reverse position, connecting automatic advance line **266** to 2nd (minus) ratio change feed line **358**. The 1st (plus) ratio change feed line **361** is exhausted via 2-3 return line **341** and return signal control valve **332**. On throttle advance, the advance signal in 2nd (minus) ratio change feed line **358** is connected, via 2nd (minus) shuttle valve **562** and 2nd (minus) actuator line **558**, to move actuator **543** in minus direction to the reverse (R) position (FIG. 3B). With forward signal line **288** exhausted, stop member **553** is inactive. Actuator **543** moves pump servo valve **563** to supply 2nd (minus) pump servo **538** to move pump ring **370** to maximum negative limit displacement (P−) position (FIGS. 3A and 4) for maximum speed ratio reverse drive. When

pump ring **370** has maximum negative displacement position, the return signal is connected—via zero stroke bias valve **363**, 1st forward return line **384**, 1st (plus) shuttle valve **561**, and 1st actuator line **557** to actuator **543**, to move actuator **543** toward reverse bias position (RB). Thus in reverse range, pump displacement and thus reverse speed ratio drive are controlled so engine speed equals the set speed of ratio governor **162**.

In reverse ratio range, actuator **543** moves motor servo valve **576** through a range of reverse range positions, always connecting control pressure line **564** to 2nd (minus) motor servo **540** to hold motor ring **401** in maximum negative limit displacement position (M−). Since motor ring **401** is not moved to intermediate negative displacement position (MI−) or 1-2-1 shift position, 1-2-1 shift signal valve **396** does not supply 1-2-1 shift signal line **402**, and ratio shift signal valve **257** cannot supply 1-2 shift signal line **489**, so 1-2 shift valve **491** cannot be upshifted. As in forward range, the downshifted 1-2 shift valve **491** engages 1st brake **65** for 1st ratio range drive and exhausts 2-3 feed line **283**, so 2nd and 3rd clutches **88** and **72** are not engaged regardless of the position of 2-3 shift valve **492**.

Thus, although pump ring **370** passes through the 2-3-2 shift point so 2-3-2 shift signal valve **386** supplies 2-3-2 shift signal line **391** and ratio shift signal valve **257** supplies 2-3 or 3-2 shift signal line **487** or **486** to shift 2-3 shift valve **492**, this has no effect and only 1st brake **65** is engaged. With 1st brake **65** engaged and propulsion hydrostatic drive control **133** providing negative pump (P−) and motor (M−) displacement, reverse drive range is established.

STEERING

There is no steering in neutral manual range drive as disconnect clutch valve **125** disengages disconnect clutch **24**, so there is no propulsion or steer drive, and supplies steer signal line **326** to positively close steer signal valve **166** (FIG. 3D), so no pressure is supplied to steer override link **704** so it cannot transmit steer request.

In pivot steer, forward, and reverse manual ranges, clutch disconnect valve **125** engages disconnect clutch **24** and exhausts steer signal line **326** to condition steer signal valve **166** for steer operation. When input speed is sufficient so steer drive will not stall the engine, input governor pressure supplied by input or 1st governor pressure line **165** shifts steer signal valve **166** against the spring **736** to the steer or open position. Then if steer hydrostatic transmission **89** has proper charging pressure, 2nd or steer control pressure regulator valve **139** supplies mainline **112** pressure via link feed line **699** via steer signal valve **166** in the open position and link supply line **741**, to steer override link **704**. When supplied with pressure, steer override link **704** transmits steer request in either 1st or 2nd direction from steer request linkage **197'** to steer servo valve **702**. In response to steer request, steer servo valve **702** is moved in a 1st or 2nd steer drive direction, and steer control pressure line **693** is connected—via 1st or 2nd steer feed line **708** or **709**, steer stroke limiter valve **592**, and 1st or 2nd steer supply line **711** or **712**, to 1st or 2nd steer servos **664** or **665**, to move steer pump ring **662** from null or zero displacement position (SN) in a 1st or an opposite 2nd steer drive direction, to 1st (1st S) or 2nd (2nd S) limit steer position for steer drive in opposite directions.

VEHICLE BRAKE OPERATION

The brake apply valve **771** of vehicle service brake system **141** supplies brake apply pressure proportional to brake demand, via brake apply line **794**, to apply right and left output vehicle brakes **37** and **37'**, and to 1st range brake signal valve **772** (FIG. 3E). Brake apply valve **771** also, in response to minimum-to-maximum brake demand, supplies mainline **112** pressure to brake-ON signal line **797** which acts to open brake signal valve **772** to connect brake apply line **794** to vehicle brake apply signal line **184** to provide a brake apply signal which is connected, via power signal shuttle valve **186**, to ratio governor **162** to increase the set speed in proportion to brake demand (FIG. 3C).

Thus when the throttle **194** is returned to idle which would decrease the set speed, the brake apply signal pressure maintains or increases the set speed to higher speed values so the propulsion hydrostatic drive control **133** provides higher speed ratios so the output drives the engine at higher speeds for increased engine braking.

In 1st ratio range, 1st brake apply pressure supplied by 1st apply line **239** to brake signal valve **772** places brake signal valve **772** in closed position shown, venting vehicle brake apply signal line **184**. Thus brake demand does not control the set speed of ratio governor **162** to control hydrostatic ratio in 1st ratio range which is used in forward and reverse drives so engine speed smoothly decreases to idle speed as the vehicle is braked to a full stop.

It will be appreciated that modifications of the above-described invention may be made.

The embodiments of the invention for which an exclusive property or privilege is claimed are defined as follows:

1. A control for use with a cross drive transmission having an input, two outputs, and output drive means for driving said two outputs; hydro-mechanical transmission means connecting said input to drive said output drive means and including hydrostatic propulsion transmission means having hydrostatic units with propulsion displacement change means and connecting propulsion kidney passages for variable speed ratio hydrostatic propulsion drive, and ratio-establishing means selectively operable for establishing a low gear ratio and a high gear ratio coordinated with said variable speed ratio hydrostatic propulsion drive for providing continuously variable overall speed ratio propulsion drive from said input to said output drive means and said two outputs; and hydrostatic steer drive means having hydrostatic units with steer displacement change means and connecting steer kidney passages for variable speed ratio steer drive proportional to steer request; said control comprising source means for providing fluid pressure; torque demand means for providing a torque demand signal proportional to torque demand; steer control means connected to said steer displacement change means including hydromechanical link means operative when supplied with fluid pressure for providing a steer request signal to, and for controlling said steer displacement change means for said variable speed ratio steer drive for variable steer proportional to steer request; ratio governor means operatively connected with said input, said torque demand means, said steer control means, and said source means and having set speed control means for providing a set speed increasing to the higher value of set speed values

increasing with both torque demand and steer request, and including signal means for providing an advance signal when input speed exceeds said set speed and a return signal when input speed is less than said set speed; displacement control means operatively connected with said ratio governor means for controlling the displacement of said propulsion displacement change means in response to said advance and return signals to respectively increase and decrease said speed ratio of said hydrostatic propulsion drive; shift signal means operatively connected to said source means, said ratio governor means, and said propulsion displacement change means for distributing shift signals at a predetermined displacement of said propulsion displacement change means; shift valve means operatively connected to said shift signal means and to said ratio-establishing means for providing selective interchanging establishment of said ratio-establishing means in response to said shift signals and for controlling said displacement control means to provide coordinated speed ratio change of said hydrostatic propulsion drive for overall continuous ratio change of said propulsion drive; steer disabling means operatively connected to said steer control means and to the one of said steer kidney passages having the lower steer kidney pressure to normally supply fluid pressure to said hydromechanical link means of said steer control means and to discontinue supply of fluid pressure to said hydromechanical link means in response to a predetermined abnormally low value of of said lower steer kidney pressure to discontinue steering.

2. The invention defined in claim 1, including input governor means connected to said input for providing an input speed signal proportional to the speed of said input; and said steer disabling means also being connected to said input governor means and being responsive to said input speed signal indicating a predetermined low input speed to discontinue said fluid pressure supply to said hydromechanical link means to discontinue steering.

3. The invention defined in claim 1, including manual control means connected to said shift valve means, said displacement control means, and said steer disabling means, and operative in a neutral position to over-control said shift valve means to prevent establishing low and high gear ratios, to over-control said displacement control means to provide substantially zero displacement, and to over-control said steer disabling means to discontinue said fluid pressure supply to said hydromechanical link means to disable said steer control means.

4. The invention defined in claim 1, including overpressure signal means connected to said propulsion kidney passages, said displacement control means, and said shift valve means, and being inoperative in low gear ratio and operative in high gear ratio for providing an overpressure signal in response to overpressure of the higher pressure in one of said propulsion kidney passages; and disabling means connected to said overpressure signal means and to said shift valve means operative to disabling position in response to said overpressure signal for over-controlling said shift valve means to prevent establishment of said low and high gear ratios to disestablish said propulsion drive.

5. The invention defined in claim 4, wherein said disabling means is connected to said ratio governor means to, in said disabling position, over-control said ratio governor means to cut off said advance signal and continuously provide said return signal in response to said overpressure signal to decrease speed ratio.

6. The invention defined in claim 4, including output governor means connected to said output drive means for providing an output speed signal proportional to the speed of said output drive means; zero stroke signal means operatively connected to said propulsion change means for providing a zero stroke signal in response to substantially zero displacement; lock means operatively connected to said disabling means operative after said overpressure signal moves said disabling means to said disabling position to retain said disabling means in said disabling position until said output speed signal indicates low output speed and said zero stroke signal indicates substantially zero displacement.

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